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THE JOURNAL

of

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

SEPTEMBER 1912

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ANNUAL MEETING: NEW YORK, DECEMBER 3-6

SOME INTERESTING DATA FOR OUR MEMBERSHIP

A very conservative estimate, based upon reliable data, shows there are in this country at least 10,000 leading engineers and directors of industries capable of satisfying the high requirements for membership in this Society. Therefore, less than half of those eligible are supporting the Society in the furtherance of the objects for which it was founded—"to promote the arts and sciences connected with engineering and mechanical construction."

With the added strength and support of these 5000 engineers now outside its ranks, the Society could accomplish much that cannot now be undertaken.

This large addition to our membership must be made very carefully and deliberately, as the present standard of membership must by all means be maintained. The task of increasing the membership is entirely in the hands of those now enrolled. They must choose those men worthy of the Society's recognition, they must approach them and explain how their membership in the Society will be mutually advantageous.

This Committee wishes to point out the necessity for prompt and earnest coöperation in this important movement, and believes that *all* of the members will cheerfully coöperate when they realize the necessity for such action.

COMMITTEE ON INCREASE OF MEMBERSHIP

I. E. Moultrop, *Chmn.*
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SEPTEMBER 1912 VOL. 34 No. 9

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OF

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MEETING IN GERMANY

For the more complete arranging of the joint meeting of the Society with the Verein deutscher Ingenieure, to be held in Leipzig, June 23-25, 1913, Dr. Conrad Matschoss, Dozent of the Royal Polytechnic High School of Berlin, is now in America representing the German society, and tentative plans will soon be announced. From the details thus far known it can be foretold that a most unusual opportunity will be afforded to visit the various German industries and to observe the progress made along engineering lines in that country. The textile district around Leipzig, the machine tool and various other industries at Düsseldorf, the Allegemeine Electricitäts Gesellschaft at Berlin, the largest manufacturers of electrical apparatus in the world, Siemens Schuckert, Ludwig Loewe, who entertained members of the Society during the visit of 1889, and the chemical works in Frankfort, are being considered as points of interest to be included in the program. Special opportunities will also be provided for a comprehensive study of the great German Museum of the Technical Arts and Industries at Munich. There will also be a trip on the Rhine and receptions in several of the cities, and visits to the various shipyards and to the art galleries in Dresden and other cities. In addition to these official excursions, invitations are being received by the Society from a number of prominent German firms which offer their hospitality to our members and show the great interest which is felt in the trip.

The Committee of Arrangements, Col. E. D. Meier, Chairman, is assured of adequate steamer accomodations, affording the maximum of comfort with the minimum of expense, provided the party is large enough. It is desired to charter a steamer, which will make possible the gratifying of all tastes. Guests will be welcomed and may be included in the party.

In addition to his mission in connection with the Joint Meeting, Dr. Matschoss is charged by the Verein deutscher Ingenieure to make a tour of America to study our educational methods and to report on these in connection with the part they are taking in the modification of the curriculum in Germany so as to produce technical graduates who are more immediately available in the shop. The Council and the Committee of Arrangements for the Joint Meeting have in charge an itinerary for Dr. Matschoss which will enable him to secure the information he seeks, and members of the Society in the various cities which he may visit are urged to coöperate by placing facilities for securing such data at his disposal.

CURRENT AFFAIRS OF THE SOCIETY

The opportunities for serving the profession of engineering were never greater than they are at the present time. The Society is seeking to enlist in this service the interest of the most eminent engineers in every branch to contribute of their knowledge either papers or discussions, and members should extend to men of attainment invitations to become identified with the Society.

All suggestions along such lines are being actively followed up by the Committee on Increase of Membership, composed of members of the Council and other members of the Society, and the coöperation of the membership generally is encouraged. Accompanying this increased activity, however, is the increasing strictness of the Membership Committee in its interpretation of the requirements for admission. The membership can greatly assist by explaining to applicants the essential requirements for admission and the necessary care which must at all times be enforced.

VISIT OF DR. UNWIN

Dr. William Cawthorne Unwin, Honorary Member of the Society and President of the Institution of Civil Engineers of Great Britain, is expected to arrive in this country the latter part of August, and will be welcomed by many members of the Society.

WILLIAM MARSH RICE INSTITUTE

The Society will be officially represented at the opening of the William Marsh Rice Institute in Houston, Texas, on October 10-12, by its President, Dr. Alexander Crombie Humphreys, president of the Stevens Institute of Technology. The new institute is destined to become one of the principal institutions of the South and men of letters and science from all countries of the world will gather to celebrate its opening.

INTERNATIONAL CONGRESS FOR TESTING MATERIALS

The members of the Society and ladies are cordially invited to attend the joint reception given in honor of the delegates to the International Congress for Testing Materials by the American Society for Testing Materials and the three founder societies, in the Engineering Societies Building, New York, on the evening of September 2. No further notice of this meeting will be made to the membership, but it is anticipated that a large audience will assemble to greet the delegates.

Calvin W. Rice, *Secretary*

MEETING OF THE SUB-COMMITTEES ON THE INDUSTRIES

A meeting of the newly appointed sub-committees of the Committee on Meetings, including committees on Textiles. Administration, Cement Manufacture, Machine Shop Practice. Iron and Steel, Air Machinery, Railroads, Hoisting and Conveying, Industrial Building. and Fire Protection, was held in the Engineering Societies Building on July 29, and was very largely attended. Dr. Alex. C. Humphreys, President, and Calvin W. Rice. Secretary, were present, and the Committee on Meetings was represented by H. deB. Parsons and Dr. Charles E. Lucke. Chairman.

Dr. Lucke, who acted as Chairman of the meeting. outlined its general purpose, stating that the Society was now making an experiment, the result of which rested with the sub-committees and the individual members. In addition to the meetings held for the discussion of strictly professional matters, directed for the benefit of the members, it is proposed to bring out the broad problems common to many or all of the different industries, applying to their solution the methods adopted by the most successful. The emphasis will, of course, be laid on those problems which properly fall within the province of engineers, rather than of traders, lawyers, or financiers.

For the working out of this experiment, groups of men have been appointed, known as committees, charged with the responsibility of bringing before the Society both the problems and their successful solution, each group being invited to coöperate with the other in any presentations which may be made. Specialists on the various subjects have been chosen and the subjects will be treated in such a way as to be helpful to all. To conform with the professional ideals of the Society, these presentations must represent scientific advance and at the same time be of practical benefit. If possible whole sessions of the Annual Meeting will be assigned to the sub-committees, for a complete discussion of the problems presented.

A general discussion followed Dr. Lucke's remarks.

The Committee on Cement Manufacture, F. W. Kelley, chairman, has already chosen as its leading topic for the Cement Session of the Annual Meeting, Depreciation and Obsolescence, and a letter has been issued to other sub-committees requesting statistics as to the obsolescence of all types of apparatus. The following are specifically mentioned: electric generators, steam engines, turbines and auxiliaries: piping, pipe fittings, steam boilers, boiler auxiliaries, stokers and coal handling equipment, electric lines and line apparatus, electric motors, lineshafts and pulleys. A second topic which has been suggested, Prevention of Accidents to Employees, also demands coöperation from other committees, and data have been asked for upon the protection of gears, pulleys, shafting, electric wiring, overhead runways and platforms, and in general the methods of securing the attention and coöperation of the workers themselves in conjunction with mechanical means for preventing accidents to them. An invitation to coöperate on these topics is extended to the membership at large.

THE PRINCIPLES OF VALUING PROPERTY

WITH SPECIAL REFERENCE TO INDUSTRIAL ENTERPRISES

By Henry K. Rowell

ABSTRACT OF PAPER

The purpose of the paper is to bring out some of the fundamental principles involved in the valuation of property. The distinction is defined between the word value, and a valuable consideration, and the several purposes for which value is to be determined, each of which may vary from the other and still be within the truth. The commercial value is the one most essential to an industry and should be the determining factor for purposes of tax, insurance or sale.

No valuation can be properly ascertained without accounting for the depreciation.

Capitalization is a delicate and intricate question to deal with. The author's experience is that most if not all writers skip over this subject lightly, and he has yet to find a real logical discussion of the subject.

In all mechanical and industrial processes the same laws of growth and development are as potent as are the laws of evolution in the natural growth and development of the physical world.

THE PRINCIPLES OF VALUING PROPERTY

WITH SPECIAL REFERENCE TO INDUSTRIAL ENTERPRISES

By Henry K. Rowell, Boston, Mass.

Member of the Society

In general every industry should be revalued at stated periods. The laws of some states require a revaluation every five years, or for purposes of assessment or insurance every seven years. In this paper the leading principles laid down for defining and ascertaining the value of property for various purposes are concisely expressed.

2 In discussing the several kinds of values, reference is made to a number of cases handed down from the Supreme Courts of several states and the United States, and some of the cases establishing the precedent upon which the rules of law are founded may be found in Judson's Taxation. In some cases the language used is that of the reported cases and in others of the textbook. The matter contained in the discussions of depreciation and capitalization is out of the author's own experience, with a brief reference to the report of the Tariff Board on the worsted and woolen industry.

DEFINITION OF TERMS

3 Following is a definition of the terms used, or referred to:

a *Cost of plant* includes the cost of the materials, labor, professional services required to produce the physical property, interest and insurance during construction, and the amount required to put the plant in operation.

b *Cost of operating* includes labor, supplies and incidentals necessary to manufacture the product, but does not include the stock in process, repairs and maintenance.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

c *Fixed charges* includes taxes, insurance, interest on current liabilities, and bonds.

d *Overhead charges* includes oversight, office and counting room expenses, advertising and selling.

e *Depreciation annuity*, the sum set aside as a sinking fund to replace the plant.

f *Annuity*, the sum set aside to discharge the funded debt at the end of a stated period.

g *Amortization*, the annuity set aside as a sinking fund to redeem the capital invested at the end of a stated period. The term refers particularly to the death of an industry from its exhaustion, as a mine or quarry. It is not depreciation, although analogous to it. Depreciation infers a continuance by renewals to the tangible property. The term amortization is not again referred to in the following discussions.

VALUE OF PROPERTY

4 In determining the value of property all the factors enumerated enter into the problem, and each must be given its proper amount of consideration and correct application to bear the test of logical reasoning. Every case of appraisal should be approached with the idea that the results may have to stand the test of the court.

5 Value does not mean cost, nor does it necessarily mean the purchase nor the selling price. It means the desirability or worth of one thing as compared with that of another, with reference to its usefulness. Value as defined by law is measured in terms of money.

6 A valuable consideration, however, within the meaning of the law may consist either in some right, interest, profit or benefit accruing to one party, or some forbearance, detriment, loss or responsibility given, suffered or undertaken by the other. It is under this definition that a court of equity would compel the specific performance of a contract where damages would not give adequate compensation as in a case where the thing contracted for is specific and its counterpart can not be purchased in the open market.

7 The rule of law would be to place a value upon the loss or non-performance of the promise and award the damages in a payment of money. Equity follows the law wherever possible,

and in a suit for damages where compensation can be given by the payment of a sum of money determined by the value of the property in question, equity would not grant specific performance, nor what is termed a payment in kind.

8 The theory of the law is that money as the universally recognized medium of exchange and measure of value, is always acceptable and leaves the one who rightfully possesses it free to purchase what he wishes to satisfy his requirements or pleasure. It is very rare, indeed, that a demand or pleasure can not be satisfied by the purchase of something else.

9 Another reason for preferring a payment of damages in money is that upon the payment of money litigation ceases; whereas, if specific performance, or payment in kind were granted, the courts would have the difficulty of enforcing the specific performance, or payment in kind.

10 Payment in kind, in the case of water rights, has been allowed in England, by act of parliament.

TAX VALUE

11 Tax value is defined by the courts and stated by the legislators of the several states to be the true or fair cash value of the property at the time of assessment. The fair cash value is the price which a willing and voluntary seller will accept from a willing purchaser who has the money to pay at the time of sale. In the assessments of taxes the rates will vary from nothing in the case of exemption to over $20 per thousand. Some localities may have a low rate of valuation with a low rate of tax, others may have both a high rate of valuation with a high rate of tax, while others may have a moderate rate for both valuation and tax. There is no violation of the law as long as the rates are uniform and equal. The United States Supreme Court laid down the rule that uniform and equal refers to geographical uniformity where the rates operate generally throughout the United States, and uniformly wherever the subject of the tax is found. The court has declined to define the word equal as no rule can be formulated to cover every case, but upon the general question it is said that equal protection of the law means that no person or class of persons shall be denied the same protection of the laws enjoyed by other persons or other classes of persons in the same place and in like circumstances.

12 Some of the states hold that equal and uniform mean in-

trinsic equality and uniformity, which is not apparent equality or uniformity. The Supreme Court of Massachusetts said that a fair cash value is irrespective of value placed by the assessors upon similar property in the same city, the question being whether it had been valued at more or less than its fair cash value, and not relatively more or less than similar property of other persons.

13 The amount of taxes assessed is usually taken from about two-thirds to three-quarters of the full value of the property, but this is not always the case. While making an appraisal recently the author found that the rule of the locality was to collect the amount of tax upon 80 per cent of the full value, while the tax rate was $18 per thousand upon a fair valuation.

14 It is obviously immaterial what the basis of valuation is if it is uniform as to all property within the territory, or as to the class of subjects upon which the tax is laid. This is recognized in the requirements of state constitutions that taxation shall be uniform upon the same class of subjects within the territorial limits of the authority imposing it. Thus, if all the property in the state were valued on the same basis it would be immaterial to the individual tax payer whether he paid 1 per cent on a valuation of $1, or 2 per cent on a valuation of 50 cents. If there was no general property tax levied by the state, based upon valuation, it would make no difference whether property in one town or county was valued on a higher basis than property in another. But within the territory wherein the tax is levied, inequality of taxation results as certainly from inequality of valuation as from inequality in tax rate. The failure to recognize this fundamental principle in taxation can result only in misleading comparisons.

15 It has been held by the courts that the evidence of the valuation of other properties is admissible as a possible assistance in determining the cash value of the property in question, when the cash value and not the proportionate value is the point in issue.

16 Stock market quotations of the company's securities may be considered because the stock and indebtedness represent the property. But they are not to be regarded as conclusive standards or tests of value. In the language of the United States Supreme Court such quotations represent the faith which a purchaser of stock in such a company has in the ability with which

the company will be managed, and in the capacity to make future earnings. Whether it is well or ill founded, is but a matter of opinion. While the value of the property is one of the material facts going to make up the market value of the stock, it is not the sole one. Speculation has been known to exercise a potent influence on the market price of stock. But assessors can use this as evidence of value.

17 Capitalized net earnings as a means to determine the fair cash value of public utility, or quasi public corporations, including the stock, franchise and tangible property, has been allowed by the court. The basis of the valuation was taken as the net earnings for the year, which did not appear to be an exceptional one, and capitalized at 6 per cent; this ratio being adopted in a special case, is the rate adopted in states where assessments were made upon the basis of net earnings.

18 The court also said that this rate was less than some advanced advocates of municipal ownership were willing to guarantee to investors in securities of this character, but it was deemed justifiable by the considerations that attended the real investor's purchase of stock. To the value of the stock and franchises determined by capitalization of net earnings was added the value of the tangible property. In a case where there were no net earnings, but an apparent loss, the court said there was no basis upon which to assess the stock above the value of the tangible property.

INSURANCE VALUE

19 In cases of personal property the measure of damages for loss of the insured property is its market value at the time and place where the loss occurred, within the amount named in the policy. In the case of damage to an insured building, the rule is indemnification to the owner, not exceeding the sum insured. The question is not what someone would have paid for the building, but what amount will indemnify the insured for the loss sustained.

20 Where goods are insured the insurer is bound to pay their value at the time of loss, where damaged only he is bound to pay the difference between the value in the sound and damaged condition. Market value, and not local or peculiar value must control.

21 In the case of the destruction of a building, where the policy

is not valued, the damage would be the value of the property as it stood upon the ground the day it was destroyed. If the building was old and dilapidated by use and decay, its value in that condition should be the ground of recovery. If the policy stipulates that the cash value of the property destroyed or damaged shall not exceed the cost of rebuilding it, and in case of damages from use or otherwise that a suitable deduction shall be made from the cost of repair, the measure of damages is the cost of repairs, if thereby the property is rendered as valuable as before. If less valuable than before, then the difference must be added to the cost; if more valuable, it must be deducted.

22 Under the laws of Massachusetts, no insurance company shall knowingly issue any fire insurance policy upon property within the Commonwealth for an amount which, with any existing insurance thereon, exceeds the fair cash value of the property, nor for a longer term than seven years. If the buildings insured against loss by fire are totally destroyed by fire, the company shall not be liable beyond the actual value of the insured property at the time of the loss or damage.

FAIR CASH VALUE

23 Fair cash value is defined as being the price a willing seller would accept from a willing purchaser, without any undue advantage to either party.

24 The influences which affect the salable values of property are variable and often complicated. It has been said that differences between assessors on questions of valuation of the same class of property are no greater than frequently arise between witnesses in a trial on questions of value. There is no certain, definite standard of values excepting that of money and standard marketable articles. Many influences, tangible and intangible, affect the salable value of property, real and personal, both in city and country, so as to make its real valuation a work of great difficulty, resulting in inevitable inequalities. Mere mistake in judgment is no discrimination.

25 The Massachusetts rule is that the fair cash value is determined upon the basis of a willing seller and a willing purchaser.

26 In New Jersey the rule has been stated as being the ordinary selling and buying price for cash at the time, irrespective of the actual cost of the property, the amount it would realize at a public sale being a good criterion. This rule would seem to

admit of the possibility of some undue advantage to the purchaser, as a public sale might not always be a willing sale on the part of the vendor. On the other hand, the reason for the rule suggesting public sale as a criterion for establishing values is undoubtedly based upon the possibility of inflating values for some reasons, or discounting them for other reasons; also for the purpose of lending aid to a uniform and equal basis of valuation on the ground that the records of such sales of property of the same class, within the territorial limits, will be available to form a standard of comparison.

27 The Massachusetts rule employing the words willing seller and willing purchaser implies the status of equality between the parties which is not in the nature of a forced sale. In a case where a manufacturing property was sold at public auction, on account of the owner's personal wish to sell the property, there is the fact of a willing sale to the purchasers, who paid about one-quarter of the appraised value of the property, as tangible property. If the sale had been made under legal process, by order of the court, it would have been a forced sale, even though the purchase price paid was the same. The price paid under the circumstances may be held as not representing the fair cash value. The sale can be said to more closely resemble a sacrifice by the owner.

COMMERCIAL VALUE

28 The commercial value is the estimated worth of property placed upon it as a business proposition, the chief considerations being the earning capacity of the plant and the amount of profit to be derived from it, both present and prospective. Involved in these considerations are location, facilities for receiving and shipping goods, available supply of help, management and reputation.

29 This value is to be determined by a proper capitalization of the earnings, which for manufacturing properties will fairly represent the market value of the property. It is evident that, as to the owners of a prosperous industry, the commercial value may be above the market value of property of the same class, also that some industry may be below the market value of property of the same class. The method of management may have the weight of influence to determine the commercial value in some cases, or a peculiarly favorable market for the product. These reasons are largely of local value rather than of general value. In the case

of poor management there is a factor that can usually be remedied. In the case of a peculiarly favorable market, the influence of competition, or the changes in fashion, will in time level the particular advantages accruing to the individual plant.

30 Industries may be so located as to involve the expenditure of a larger amount of money in the establishment of their tangible property than other plants more favorably located, owing to some particular advantage gained at the time of their inception. With the advent of new inventions and improved methods, the reason for having the plant in such a locality has passed away. Conditions and environment may result in larger operating expenses reducing the profit. As a business proposition the value of the tangible property can be no greater than that of other plants of the same class more favorably located and costing less; and their commercial value should be determined by an amount equivalent to the charge upon the profits owing to the unfavorable conditions. This is illustrated by many cases where plants have been located in out-of-the-way places to take advantage of some water power. With the advent and increased use of electricity, with the growth of the plant beyond the capacity of the water power, with a growing scarcity of help, and with a more favorable locus for producing the same kind of goods in other places, these plants have been forced to suffer loss and prestige.

31 In valuing industrial property for tax assessment, insurance indemnity or for sale, the commercial value is the leading consideration, as upon it would depend the attitude of the purchaser towards the seller. In determining the value of property upon the basis of net earnings or profits the amount to charge off against the tangible property would be the extra expenses involved in manufacturing the product that could be saved if the plant was working under favorable circumstances. This would not necessarily affect the value of the collateral intangible or incorporeal property, except in so far as the loss of profit might affect the sale of the stock, which might also be affected by loss of profit due to poor management. While poor management may have a very material affect upon the commercial value of an industry, it is subject to correction, and the result of poor management may be felt for some time after the cause has been removed. It is not inherent in the property the same as unfavorable location, the betterment of which would mean the removal of the plant.

32 In valuing the tangible property, its adaptability for other uses is to be considered. This involves the amount of capital which must be expended to put the property into a proper condition to carry on the business for which it may be purchased. In the event of carrying on the same class of business the question of the economical organization of the plant, as compared with other well arranged plants entering into competition would be involved.

33 The uniform and equal valuation of textile, and other industrial property and power plants should by this time be well within the range of possibility, and no one should be any better able to make fair and unprejudiced appraisals than the mechanical or industrial engineer. There is a sufficient amount of information at hand true to the facts of cost, both for installing plants and for operating them, covering all classes, especially of the textile industries. The engineer, who has made certain industries a specialty in all of the processes of mechanical organization, cost of establishing, and cost of operation, should be better qualified than any other man to act as an expert in such valuations. He has a broad field from which to gather his information and a larger opportunity to view the various grades of the same classes of industry. The ethics of his profession should lead him to be an unprejudiced witness before the court. The theory upon which the expert witness is allowed to testify is that he can assist the court to arrive at a just decision between the parties in a controversy, owing to his superior knowledge and experiences. His evidence should be free from the taint of partisanship, and he should stand before the court in the same attitude as an attorney, representing his client, but serving the court. The expert witness is the only one allowed to give opinion evidence, on the theory of his special qualification.

34 No valuation can be made without taking into account the depreciation.

DEPRECIATION

35 Depreciation is the falling off in value. Physical depreciation refers particularly to the lessening value of material things, due to age and wear, and improvements in the mechanical arts. Everything of substance has some intrinsic value, however small.

36 There are several methods of computing depreciation;

some are represented diagrammatically by curved lines, others by straight lines, all of which methods show some variations, but any method adopted may be only approximately correct. Depreciation begins at once, however small, and the actual process will not proceed very far before the work of maintenance begins. Whatever the method or rate of computing the depreciation, if it is sufficient to meet the conditions its relative accuracy is near enough to the truth for all practical purposes. Then it is logical to utilize the simplest form of diagram to represent the loss, which is the straight line computed on the original cost, or the replacement cost if new at the time of making a valuation of property.

37 While it is not the purpose to discuss the several methods of computing the depreciation here, as these discussions may be read elsewhere, the rate of depreciation may be briefly considered.

38 While the rate of depreciation will vary with materials and circumstances, it may be possible to determine a fairly constant rate applicable to property of the same class, used under similar conditions and subject to the same general care and maintenance. In the case of the Chicago Union Traction Company vs. State Board of Equalization, 114 Fed. Rep. 557, it was held that an annual reduction equal to 6 per cent of the current value of cars, tracks and machinery was not excessive as the allowance for current depreciation.

39 In the textile industries, plants may be grouped together in classes, each class being subject to the same economic conditions. In the cotton industry, plants may be classified as duck mills, course sheeting mills based on a standard weight of cloth containing three yards to the pound, medium weight goods mills based on a standard of four yards of cloth to the pound, print goods mills, making goods for the general household trade, and fine goods mills. The mills of each class enter the market to sell their product in competition with the other mills of the same class. Each mill of the same class must strive for the highest efficiency in order to compete successfully with the others, and these conditions have led to a highly scientific development of the industry. In the woolen and worsted trade, the same highly efficient physical organization is being wrought out, until it seems as if there can not be any very radical changes in the methods or general types of machines used in the process of manufacture.

40 In all mechanical and industrial processes the same laws of

growth and development are as potent as are the laws of evolution in the natural growth and development of the physical world. The very nature of the materials we have had to convert into a marketable product for our use has indicated the lines of development for the mechanisms of our industrial processes. Many improvements are yet to be made and perfected.

41 One of the most notable developments and advancements in the art in recent years is the advent and successful operation of the steam turbine, and prior to it the introduction and development of electricity for all purposes. One of the greatest developments yet to come is the method of generating heat without coal.

42 A statement was recently made that no great advancement has been made in the cotton industry for about forty years. This is a cause for surprise and some inquiry. The period from 1870 to 1880 appears to mark the beginning in improved modern buildings, better lighted rooms, better fire resisting construction, more economical arrangements of plants generally. The decade from 1880 to 1890 appears to have been the period of increase of the product of machinery and of a higher efficiency of the plant generally. The decade from 1890 to 1900 appears to have been the time for introducing better organized plants, of giving more attention to the welfare of the employees and tending to raise the standard of quality generally. The last decade has witnessed the introduction and successful operation of electricity generally, better appointed power plants, more efficient machinery and buildings, and the study into the question of more efficient work people.

43 In the decade from 1870 to 1880 the average speed of the front roll of spinning frames on print goods numbers was 64 r.p.m., the average product being about 0.80 lb. in 60 hours. Today the average speed of the front roll for the same numbers of yarn is 122 r.p.m., with the average product of 1.44 lb. per spindle in 60 hours, an average increase of 80 per cent per spindle, with an infinitely better quality of yarn. The product of looms has increased from an average of about 27 lb. per week to an average of 44 lb. per week on print goods. Forty years ago the average product per week gross floor area was about 1 lb. of cloth to 7.25 sq. ft., while today in modern mills the average product is about 1 lb. of cloth per week to 3.5 sq. ft. for a week of 60 hours. These results have been calculated from the records of a number of mills of the earlier period and compared with what is seen in practice today. The increased product of machinery may show

a greater ratio than the increase per square foot of floor area for the reason that in our modern mills more alley room is allowed around the machinery, spinning frames are wider and built to a larger gage than in the earlier days, looms are larger and heavier for the same width of cloth. There is a saving in the card room, as the revolving flat cards used today occupy less room than the double system of flat top cards used years ago, and a large increase in weaving, due to the use of automatic looms.

44 Within the last ten years there have been some marked improvements in the building of machinery. It is built more strongly, on better lines, and is more rugged. This is quite apparent in the case of some steam engines, as compared with earlier types.

45 As to the depreciation rate to charge off against machinery in the case of textile machinery, it is a common thing to find groups of the same kind of machines in a mill varying in age from fifteen to eighteen years; sometimes one sees a group of machines over twenty years old, and it is not uncommon to see single machines from thirty-six to forty years old. As a rule it would be beneficial to the organization to dispense with these old machines. The writer visited a mill sometime ago where there were some old roving frames and was told by the agent that it was more economical to let the old frames remain idle than to try to run them. It is short-sighted policy that keeps such old machines out of the junk heap.

46 By the time spinning frames reach the age of fifteen years they usually begin to require more extensive overhauling than ordinary repairs. The writer has known frames of about this age to require such thorough refitting that the standards were about all that was left of the original machine.

47 During the recent investigation by the tariff board appointed by the President of the United States, some rather interesting data were obtained regarding length of time machinery has been used in worsted and woolen mills, not necessarily indicative of the rate of depreciation of such machinery, although certain inferences may be drawn. Some of these data are given in Tables 1-5.

48 There is nothing in any of the tables given to show the age of any of the individual plants from which these lists were made up, so that no allowance can be made for new industries starting up within the last ten years, but between the ten and twenty year period there is about the same ratio for each period of five years.

and after this period the amount of machinery that has been in operation is relatively small.

49 From personal observation among a large number of textile mills the average rate of depreciation appears to follow along these general lines. Up to about fourteen years of use the machinery contains practically the original parts with which it was furnished, subject to the ordinary repairs that an economical mill will keep up. Of course accidents or some unlooked-for event

TABLE 1 YEARS IN OPERATION OF WORSTED MACHINERY, BRADFORD SYSTEM

	Scouring		Carding		Combing		Gilling		Drawing	
	No.	Per Cent	No.	Per Cent	No.	Per Cent	No.	Per Cent	No.	Per Cent
Less than 5 years...........	11	16.4	173	26.3	53	14.0	137	18.5	797	32.9
5 years but less than 10 years.	26	38.8	143	21.7	136	35.8	214	29.0	628	26.0
10 years but less than 15 years	17	25.3	76	11.5	18	4.7	68	9.2	126	5.2
15 years but less than 20 years	5	7.5	61	9.3	46	12.1	66	8.9	434	17.9
20 years but less than 25 years	5	7.5	155	23.6	43	11.3	116	15.7	92	3.8
25 years but less than 30 years	2	3.0	31	4.7	47	12.4	71	9.6	214	8.9
30 years and over..........	1	1.5	19	2.9	37	9.7	67	9.1	129	5.3
Total machines.........	67	100	658	100	380	100	757	100	12,500	100

TABLE 2 YEARS IN OPERATION OF WORSTED MACHINERY, SPINNING AND WEAVING

	Mules		Frames		Looms*	
	No.	Per Cent	No.	Per Cent	No.	Per Cent
Less than 5 years..........................	92	24.9	273	20.3	4125	33.4
5 years but less than 10 years..............	121	32.7	559	41.5	1244	10.0
10 years but less than 15 years.............	12	3.2	109	8.1	2009	16.3
15 years but less than 20 years.............	34	9.2	76	5.6	2072	16.8
20 years but less than 25 years.............	100	27.0	98	7.3	2053	16.7
25 years but less than 30 years.............	11	3.0	116	8.6	396	3.2
30 years and over..........................	0	0	115	8.6	438	3.6
Total machines.......................	370	100	1346	100	12,337	100

* Includes all looms for both worsted and woolen mills.

TABLE 3 YEARS IN OPERATION OF WOOLEN MACHINERY

	Cards		Mules	
	No.	Per Cent	No.	Per Cent
Less than 5 years	40	9.6	47	8.0
5 years but less than 10 years	62	14.8	204	34.7
10 years but less than 15 years	68	16.2	143	24.3
15 years but less than 20 years	16	3.8	72	12.3
20 years but less than 25 years	36	8.6	47	8.0
25 years but less than 30 years	32	7.6	53	9.0
30 years and over	165	39.4	22	3.7
Total machines	433	100	588	100

TABLE 4 RELATION BETWEEN DOMESTIC AND FOREIGN MADE MACHINERY, GIVING THE NUMBER OF MACHINES IN PLANTS LISTED

Country	Scouring	Cards	Combs	Gills	Drawing	Spinning	Looms
United States	37	730	57	37	273	617	9517
England	18	271	323	210	2488	1486	2713
Germany	7	90	225	90	192	195	107
France	5	0	22	32	36	6	....
Total	67	1091	627	369	2989	2204	11337

TABLE 5 AVERAGE YEARS IN OPERATION OF WORSTED MACHINERY LISTED IN TABLES 1 AND 2

Years in Operation	Average Per Cent of Total Machinery
Less than 5 years	26
5 years but less than 10 years	29
10 years but less than 15 years	10
15 years but less than 20 years	11
20 years but less than 25 years	13
25 years but less than 30 years	6
30 years and over	5

may happen to a machine to cause some extraordinary expense. From the tenth to the fifteenth year there is likely to be some very general overhauling of the machinery, renewal of vital parts, and perhaps the substitution of one or more machines in a group of similar machines. The whole organization is toned up generally. Conditions will run along for the next ten years or so under the ordinary mill care when there begins to be a very general reorganization of the plant.

50 The inference might be drawn from Tables 1-5 that the general, or average, depreciation may follow the results shown, and this does not seem an unreasonable, although it may not be the true conclusion. Assuming that the inference is approximately true, it seems that a depreciation annuity of 6 per cent on the cost of the machinery is not too much to set aside in order to provide a proper and conservative amount with which to replace textile machinery. This would allow for a partial replacement at the end of the 15 year period, and a general renewal of the plant at the end of 25 years. If the plant is in a good condition at the end of 25 years and can continue to operate for a few years longer, the gain is so much greater.

51 Industrial buildings are planned for certain specific purposes. The organization of the plant is first worked out, the departments arranged most economically, then the walls and roof put over the whole.

52 Under proper conditions, buildings will last for hundreds of years, but with factory buildings, the conditions are such that it is not expected to have a building last much beyond a certain limited time, owing to the change in methods. It is noticeable, however, that the buildings for manufacturing purposes of recent years are more substantially built than they were 50 years ago.

53 In looking over the records of existing textile industries, buildings in use which are over 50 years old are not very numerous; there is a larger number of those 40 years old, and a rapidly increasing number in comparison as the age diminishes. The old buildings are wholly unsuited for modern practice. In fact, many buildings built 25 years ago are not suitable for a proper arrangement of modern machinery.

54 With more rugged and better built machinery and stronger and better made buildings it is possible that the depreciation will not be so rapid as it has been in the past. To a very large extent

depreciation is due more to rapid advance in the mechanical arts than to age and wear. The proper maintenance of a plant can retard the depreciation due to age and wear very materially. Depreciation due to advance in the arts is, in the author's opinion, one of the essential factors to be considered in the use of reinforced concrete buildings, as this type of building is not so readily adapted to be modified or changed over to meet new conditions as is the regular mill construction type. It may be, however, that radical changes are not so likely to come in the future as they have been in the past, but this seems a short-sighted view.

55 In depreciating factory buildings, a rate of about 2 per cent a year does not seem excessive, compared with the useful life of the older type of building, while about 1 per cent a year may be large enough to allow for the depreciation of the modern buildings.

56 In making up valuations of textile plants the physical depreciation for the preparatory, spinning and weaving machinery should be taken at about 4 per cent a year, while for dye-house and wet-finishing machinery the wooden parts of machinery have to be entirely renewed on an average of every five years, so that the depreciation on these machines after three years is very nearly a constant rate for a going concern, and can be taken at about 25 per cent of the replacement cost.

CAPITALIZATION

57 To capitalize is to convert an income, annuity, etc., into a single payment or an equivalent capital sum.

58 Under the subject of capitalization we must take into account what is included in the fixed charges on the plant. It is a very easy thing in handling capitalized values not to include some element in the calculations that belongs there and so raise false values, or vice versa.

59 The fixed charges usually include taxes, insurance and interest on current liabilities. The factors subject to capitalization are the annuities for depreciation, the annuities to discharge the funded debt and a fair profit to be derived from the enterprise.

60 The rate of capitalization is not always easily determined and may vary under varying circumstances for the same class of property.

61 In making up the commercial valuation of the railway

operating property of the United States in 1904 the rates of capitalization vary from about 3½ per cent to about 10 per cent based on the net earnings of the railroads, and the total of these capital amounts represented the wealth of the railroad property in this country at that time. The whole subject is discussed in Bulletin 21, Bureau of the Census.

62 In determining the rate of capitalization of values for industrial properties, Table 6 shows the factors which may be taken. The table is based on the assumption that the economic life of buildings is 50 years, and that the annuities are invested to earn a return of 4 per cent compound interest, the buildings being about three-eighths and the machinery about five-eighths of the cost of the tangible property.

TABLE 6 FACTORS IN DETERMINING RATE OF CAPITALIZATION

	Annuities	Percentage of Contributions		
1	Buildings depreciated at 2% a year	0.25	0.25	0.25
2	Machinery depreciated at 4% a year	1.50	....	....
3	Machinery depreciated at 5% a year	....	2.10	....
4	Machinery depreciated at 6% a year	....	....	2.65
5	Annuity to discharge funded debt, say	1.12	1.12	1.12
6	Fair net profit	10.00	10.00	10.00
	Total	12.87	13.47	14.02

63 As the plant grows old and improvements are made in methods and efficiency the net profit will decrease. Assuming this decrease to follow at about the same rate as the average physical depreciation of the property, there will come a time when it will cease, unless proper precaution is taken to provide a fund large enough to maintain the property by renewals from time to time; and the time will always come when some general reorganizing and renewals are necessary. It seems justifiable to take the depreciation fund at 6 per cent per year of the cost for textile machinery, and following similar periods of usefulness to allow for not only the renewals which must be provided for in due course, but for renewals due to improved and more economical machinery.

64 A gross income of 20 per cent should be conservatively assured from a new industry not only to cover the annuity

charges and the profit expected to be realized from year to year, but also the extra expenses of establishing the plant and the risk involved in floating the enterprise.

65 In capitalizing values the rate of 10 per cent is often used. If this includes the annuity charges for depreciation, then the allowance for the net profit is only about 5 per cent. If it is assumed to include the net profit only, it is a more conservative assumption.

66 The method of capitalizing a possible saving to an industry to determine comparative values is often not correct, as the capitalized values lead to exaggerated conclusions which are wrong. What percentage of saving could be effected by comparing the costs of installation and operation between two processes should be ascertained, and the values then determined from the cost of installation. If the saving amounted to 15 per cent, the cost of making a change would be considered justifiable; if to 10 per cent, the cost of making the change would not be considered justifiable, except as modified by the question of necessity, as by the renewal of an old plant under similar conditions. A saving of 10 per cent might warrant a change in methods. Thus in ascertaining comparative values by the method of capitalizing possible savings, it would seem as logical to use a rate of 15 per cent on the basis of the above reasoning.

67 A low rate of capitalization may lead to exaggerated and wrong conclusions. It is easy to mistake the principle involved, and rate the value of tangible property on the negative side of the equation. A small variation in the rate results in large variations in the values. The values for tangible property should always appear on the positive side, as all material property has some intrinsic value, however small.

68 As an example of this principle we will assume a cotton mill equipped with 500 automatic looms and 500 plain looms. The plant is making a class of goods where automatic looms can be used to advantage. The product of the automatic looms is about 45 lb. of cloth per week per loom, that of the plain looms about 40 lb. of cloth per week per loom, or say about 1,000,000 lb. per year for the 500 plain looms.

69 By using 444 automatic looms in place of the 500 plain looms a net saving of about 1¼ cents per lb. of product can be made, or a total yearly saving of $1{,}000{,}000 \times 0.0125 = \$12{,}500$. The cost of 444 automatic looms at $168 each, including installa-

tion, would be $74,592, and if we allow a salvage value of $10 per loom for the plain looms the net cost of installing the automatic looms would be $69,592. A yearly saving of $12,500 would be about 18 per cent of the cost of installing automatic looms, so that the mill would be warranted in making the change.

70 The replacement cost of 500 plain looms, if new, would be about $30,000. If we capitalize the saving of $12,500 in favor of automatic looms at the rate of 15 per cent, the capital value equals $83,333. We have already allowed a salvage value of $5000 for the plain looms so it is evident that the only value of the capitalized saving in favor of automatic looms to be applied to the plain looms is an amount having the positive value of $5000.

71 Again, assume that the cost of installing the automatic looms is $74,592. The replacement cost of the plain looms if new is about $30,000, which depreciated for age and wear is at present $5000, with no rebate by the loom company as an inducement for taking these looms out and replacing them with automatic looms. The gross cost chargeable to installing automatic looms would be about $79,592, including the value of the plain loom, and a yearly saving of $12,500 would be about 16 per cent of this cost in favor of automatic looms, which indicates that the investment would be warranted.

72 With reference to the commercial value of the plain looms, the capital value of the saving that could be made by using automatic looms can not be applied to reduce the value of the plain looms by an amount equal to more than their depreciated replacement cost, as a positive value, or $5000.

73 One of the hardest experiences is to recuperate an old plant after it has been allowed to reach such a state of physical depreciation as to become profitless, and there has been no depreciation fund provided to take care of future contingencies. Such conditions are probably more likely to occur in privately owned industries than in corporations. There are numbers of cases where plants in such a condition have been taken over and operated, but this means the sinking of new capital.

74 A case coming under the author's observation is that of an old mill which had reached the limit of its usefulness as a producer. Under a proper management, the old buildings were replaced by a substantial modern mill. During a period of depression the conclusions as to the wisdom of building a new mill

were that with the old buildings the plant would have been forced to close, owing to the stress of circumstances. The light was so much better in the new mill that the goods formerly passing as first quality in the old mill were not equal to the second quality in the new mill. Owing to trade conditions, it had been necessary to cut wages of the operatives twice, yet with the reduced wages the best operatives were able to earn more money in the new mill than they had been able to earn in the old mill before any reduction in wages had been made.

METHOD OF VALUING A PLANT

75 In the event of making a valuation of property, the first essential thing is for the appraiser to go over the property as a whole, to see how the plant is arranged, and how the arrangement compares with other plants making the same general kind of manufactured product. This survey will usually disclose any special features that may be unfavorable to a more economical arrangement, or where certain alterations might be made with benefit to the plant, as the organization and arrangement should be considered one of the vital features in ascertaining values.

76 It is not enough to say that the physical property of an industry can be replaced for a certain sum of money, or that it costs a certain amount and is to be valued at a certain other sum of money after allowing for a certain depreciation due to age, wear, obsolescence, etc. An investigation must be made into the working of the plant, comparing it with other plants in competition with it. Its location must also be considered.

77 Any extra cost of operation that can be eliminated by a proper reorganization must be accounted for and set off against the value. Sometimes the cost of reorganizing a portion of the plant is greater than the expense of making the change would warrant, but the poor feature will remain nevertheless a detriment to the plant and affect the value.

78 Inventories of property can be given by the owners of the plant, but these inventories should always be checked up by a person familiar with the business, who knows the process of manufacturing the goods produced, the machine used in each process, and how they should be arranged in each department, and who is capable of taking a birdseye view of the whole property and seeing it as a working whole, as well as segregating the several parts.

79 For this reason the engineer who is familiar with certain lines of industry is better able in general to take an unprejudiced as well as a more comprehensive view of the subject, owing to his wider experience and more special knowledge of engineering than a person engaged in commercial lines.

INCREASE OF BORE OF HIGH-SPEED WHEELS BY CENTRIFUGAL STRESSES

WITH FORMULAE FOR FORCE AND SHRINK FITS

By Sanford A. Moss

ABSTRACT OF PAPER

Unless proper precautions are taken, a high-speed turbine wheel expands at the hub a sufficient amount to cause an appreciable increase in the bore so as to make the wheel slightly loose upon the shaft, soon causing distress.

There is given in the paper a general discussion of stresses in a rotating wheel; a mathematical method for computation of the increase in bore of a rotating wheel due to centrifugal stresses; an investigation of the actual numerical value of the increase of bore under various practical conditions; an experiment giving an actual measurement of the increase in bore of a wheel at high speeds, showing close agreement with computed values; a description of methods of preventing damage due to the increase of bore by using large amounts of force fit; and a mathematical treatment of force fits.

The increase of circumference of the bore is proportional to the original circumference and the tangential stress and inversely as the modulus of elasticity. In the experiments the readings showed that the bore expanded at each speed an amount almost exactly equal to the computed values.

For a wheel with 12 in. bore and 30,000 lb. maximum stress, which is permissible with special grades of steel, this increase of bore comes out over 0.01 in., which gives an appreciable loosening of the shaft fit.

A complete treatment of the subject of force and shrink fits based on the maximum shear law is given for the first time so far as known.

When the hub is in place on the shaft, it is expanded by a certain amount and the shaft is also compressed by a considerable amount. The ratio of expansion of the hub to the total force fit is the same for a given wheel and shaft regardless of the amount of the force fit. A curve is given showing values for various cases. Curves are also given showing values of radial stress at the bore of the hub which is also the pressure between hub and shaft, and the maximum effective stress at the bore of the hub which is usually the greatest stress in the system. The usual case of a shaft with a comparatively small hole is practically equivalent to a solid shaft for these matters.

For an average case the hub bore expands about 0.6 of the amount of the force fit and for a force fit of one mill per inch of bore the radial stress is about

13,000 lb. per sq. in. and the effective tangential stress is about 27,000 lb. per sq. in. For other amounts of force fit the stresses are in proportion.

For the average case above mentioned the pressure to force the hub on the shaft is about 1560 lb. per in. of bore and per inch of hub length for a force fit of one mill per inch of bore.

INCREASE OF BORE OF HIGH-SPEED WHEELS BY CENTRIFUGAL STRESSES

WITH FORMULAE FOR FORCE AND SHRINK FITS

By Sanford A. Moss, West Lynn, Mass.

Member of the Society

Recent advances in high-speed steam turbines and the apparatus driven by them have opened up a great many new problems. One of these is the fact that, under certain circumstances, unless proper precautions are taken, a high-speed turbine wheel, or the like, expands at the hub a sufficient amount to cause an appreciable increase in the bore, so as to make the wheel slightly loose upon the shaft and capable of pounding back and forth. This pounding causes a mechanical increase of the looseness, so that the machine soon gives evidence of distress.

2 So far as the writer knows, the first one to direct attention to this problem, to suggest means of investigating it, and to offer a solution, was Richard H. Rice, chief engineer of the Lynn works' turbine department, General Electric Company. The present paper gives an account of work on the problem done under the general direction of Mr. Rice by various members of his department.

COMPUTATION OF INCREASE OF BORE OF A ROTATING WHEEL

NATURE OF STRESSES IN A WHEEL OR RING

3 It is the purpose first to give a general account of methods of computing stresses in circular bodies. When there is rotation this becomes a very intricate subject and details and formulae must be sought in works specially devoted to it.

4 It is assumed that the disk is thin, that is, that the thickness is negligible compared with its diameter. Then conditions are the same at all points on a line parallel to the axis. This is not

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strictly true, as the central part of a hub is strained differently than the ends. However, in most cases this effect is slight.

5 At each radius in the disk there are two stresses, one in the radial direction denoted by S_r and one in the tangential direction denoted by S_t. If we consider the effect of these stresses on the metal of a thin ring at the interior of the disk, we find the circumference of the ring to be increased mainly by virtue of the tangential stress, and the thickness mainly by the radial stress. However, when a metal is extended in one direction by a stress there is a shrinkage in a direction at right angles, the ratio of the deformations being called "Poisson's ratio," which we will denote by V. For ordinary kinds of steel this has a value of 0.3. That is, the increase of circumference of a thin ring in the interior of the disk is less than that which would be produced by the tangential stress if acting alone, by an amount which is 0.3 of the extension which would be produced by the radial stress acting alone. Similarly the increase in the thickness of the ring is less than that produced by the radial stress acting alone. In any case, the ratio of stress to extension per unit length produced by it if acting alone, is given by the modulus of elasticity E, usually taken as 29,000,000 lb. per sq. in.; or, if l is the total extension in a distance L, we have the well-known law

$$E=\frac{S}{\frac{l}{L}}\dots\dots\dots\dots[1]$$

The combination of the above principles with the laws of the centrifugal forces on the various elements of the disk gives formulae which enable computations to be made of the stresses in the disk at all points during rotation.

6 In order to determine the safety of a structure where there are stresses in more than one direction, it is necessary to find an effective stress which corresponds so far as yielding is concerned to the stresses in a specimen of the same material subject to simple tension in a testing machine. It is now almost universally agreed that Guest's maximum shear law is the proper one. This states that in cases such as the present, where there are stresses in two directions only with zero stress in the third, and where the stresses are either both tension or both compression, the greater of the two stresses is the effective stress. This effective stress is equal to a simple stress which would produce like results so far as yielding, safety, elastic limit, etc., are concerned.

7 This is not true, however, when there are stresses in all three di-

rections, as in the case with a rotating disk of appreciable thickness where there is change of stress in the axial direction; or with two stresses with opposite sign, as in the stresses in the hub due to force fits, discussed in Pars. 29-39. In such cases the maximum algebraic difference between the greatest and least of the stresses existing at a point taken with proper sign gives the effective stress or equivalent simple tension.

8 In most cases of properly designed rotating wheels with a hole at the center the maximum stress is the tangential stress at the bore. This is, therefore, the stress which is to be considered in order to ascertain if the wheel is safe and is further the stress which must be determined in order to discuss the problem of the present paper.

FORMULA FOR INCREASE OF BORE

9 As we proceed outward from the bore in a rotating wheel, the radial stress mounts up rapidly, due to the fact that the centrifugal forces of the outer portions of the disk pull outward on the inner portions. The nearer the center the less the material in the inner circle, and therefore the less resistance to radial elongation. At the bore there is no force whatever resisting radial elongation, so that the radial stress becomes zero as this inner diameter is reached. We have then on the inner wall of the cylinder forming the bore, material which is stressed only in the tangential direction and the increase of circumference is exactly the same as if we had a straight bar of the same length stressed an amount equal to the tangential stress at the bore. This is expressed by the simple relation which is the fundamental equation of our problem:

$$X = \frac{S_{to}\, D_o}{E} \qquad \text{[2]}$$

where

X = increase in diameter in inches of the bore, due to the centrifugal stresses at a given speed

S_{to} = tangential stresses at the bore in pounds per square inch at the same speed

D_o = original diameter of bore in inches

E = modulus of elasticity, 29,000,000 for steel

10 It can be shown that the stresses at any point in a given wheel or at a similarly situated point in any other wheel of similar proportions, are directly proportional to the square of the peripheral wheel speed. Hence the increase of bore in which we are interested varies directly with the square of the peripheral wheel speed with

a given wheel. In any case of the same or geometrically similar wheels, the increase per foot of bore increases directly with the square of the wheel speed.

NUMERICAL MAGNITUDE OF INCREASE OF BORE

11 The allowable values of the tangential stress at the hub are quite high, since a centrifugal load is one of the easiest loads which can be applied and is of the nature of a pure dead load in a beam. It is not possible to apply or remove a centrifugal load suddenly, but only in the most gentle manner. Also the maximum stress usually occurs only in a single set of fibers which are reinforced by adjacent fibers with decreasing stresses. There need be no "factor of ignorance" and maximum stresses can be known with as great accuracy as it is desired to give to the mathematical computations. In many cases 20,000 lb. per sq. in. is a conservative stress. In some cases with special grades of steel 30,000 lb. per sq. in. is possible. For a wheel with 12 in. bore and 30,000 lb. maximum stress the increase in bore computed from [2] is 0.0124 in., which is an appreciable

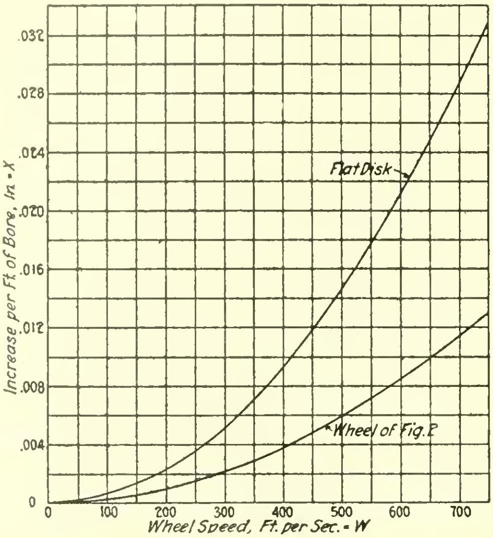

Fig. 1 Increase of Bore of a Rotating Wheel

Ratio of Diameter of Bore to Diameter of Disk = 0.353. This Ratio may be changed considerably without appreciably affecting the Results.

Approximate Dead Weight per Pound of Disk (equal in Both Cases) = 100,000 Lb. per Ft. of Disk Circumference at 534 Ft. per Sec.

amount. For other stresses and other dimensions of bore the increase of bore is in direct proportion. When the stress is 29,000 lb. per sq. in. the increase in bore is 1 mil per in. of bore.

12 Fig. 1 gives numerical values of increase in bore for various wheel speeds for two different shapes of disk. The upper curve is for a disk of uniform thickness. This shape is only possible with comparatively low wheel speeds and the stresses become excessive with speeds ordinarily used in turbine practice. The lower curve is for a disk of the more efficient shape shown in Fig. 2. In both cases the computations are made for a turbine bucket wheel with a load due to centrifugal force of buckets at 3600 r.p.m. of 100,000 lb. per ft. of circumference. The shapes of the curves are affected somewhat by the magnitude of this outer dead load. In both cases the ratio of bore to disk diameter is 0.353. The curves are not greatly affected by difference in this ratio. The curves give very nearly the values of increase in bore per foot of bore for various wheel speeds for a disk of uniform thickness, which is about as poor a shape as would ever be used, and for a disk of what is probably as efficient a shape as would ever be used, with an average bucket load. The increase in bore for any shaped disk whatever will probably lie between the two curves.

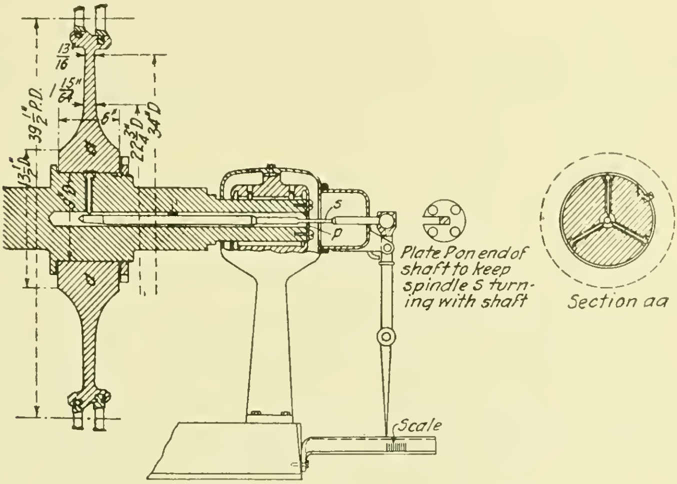
13/16
1 15"/64
39½" P.D.
6"
13½" D
22¾" D
34" D
S
P
Plate P on end of shaft to keep spindle S turning with shaft
Section aa
Scale

FIG. 2 DIAGRAM OF DEVICE FOR MEASURING THE INCREASE OF BORE DURING ROTATION

13 The curves of Fig. 1 were constructed as follows: The wheel

whose dimensions are given in Fig. 2 is the wheel of a standard steam turbine with a web diameter of 34 in. In the course of the design of this wheel the maximum stress was computed for the rated speed, 3600 r.p.m., and came out about 20,400 lb. per sq. in. From the value of this stress and formula [2], the increase in bore at the rated speed was found. The effective bore, taking account of keyways, was 12 in. From the law that the stress and hence the increase in bore varies directly with the square of the wheel speed, the lower curve of Fig. 1 was drawn, taking bore as 12 in., however.

14 Next was taken the case of the upper curve for a flat disk. From the formulae for stresses in flat disks given in Stodola's "Die Dampfturbinen," fourth edition, pp. 248 and 249, can be deduced the following formula for the tangential stress at the bore of a flat disk S_{to}, where S_{r1} is the radial stress at the circumference due to bucket load, W is the wheel speed in feet per second and R is the ratio of bore diameter to web diameter.

$$S_{to} = 0.0261W^2\ (0.7R^2+3.3)+\frac{2S_{r1}}{1-R^2}\ldots\ldots\ldots\ldots[3]$$

15 This is for a steel disk whose modulus of elasticity is 29,000,000 lb. per sq. in., Poisson's ratio 0.3 and density 0.28 lb. per cu. in.

16 The centrifugal force due to the bucket load on the previous disk, 100,000 lb. per ft. of circumference, was taken as being applied to a flat disk of about the same weight, giving $1\frac{1}{4}$ in. axial width, and 34 in. web diameter. This gives a radial stress S_{r1} of 6670 lb. per sq. in. at the circumference of the flat disk. The ratio of the diameter of bore to the web diameter R was taken as in the previous disk 0.353.

EXPERIMENTAL MEASUREMENT OF INCREASE OF BORE OF A ROTATING WHEEL

17 The magnitudes of the increase of bore in a number of actual turbine wheels, as determined by substituting computed stresses in equation [2] came out so large that it was decided to make an actual measurement of the increase in bore when a wheel was rotating.

18 Fig. 2 shows the apparatus diagrammatically and Fig. 3 the results obtained. The turbine wheel was mounted on a shaft with a light force fit. A delicate micrometer arrangement was provided so that change in bore of the wheel while it was rotating with the shaft could be observed. The wheel was gradually increased in speed and meanwhile the readings of the micrometer taken. These

showed that the bore expanded at each speed an amount almost exactly the value computed by the methods previously given. The micrometer arrangement proved very positive and there was no uncertainty regarding the measurement of the increase of bore. The maximum amount measured was about 0.007 in. The observations were accurate to about 0.0005 in.

19 The wheel was forced on the shaft with a force fit of 0.0014 so that it was loose on the shaft by about 0.0056 at maximum speed. The duration of the tests was not great enough to show any difficulty due to this.

MEASURING DEVICE

20 The measuring device shown in Fig. 2 consisted of three pistons sliding in radial holes in the shaft pushed outward by means of a tapered plug moving axially in a hole in the center of the shaft. The motion of the central plug was given by means of a lever with a ball and socket joint fulcrumed so as to give considerable multiplication and read by a pointer passing over a scale. A displacement of the pointer by 1 in. corresponded to an increase of bore of 0.004 in. The position of the pointer was certain to within about $\frac{1}{8}$ in. motion along its scale, corresponding to 0.0005 in. change of bore. While the speed was being varied an attendant kept his hand on the pointer, moving it back and forth a slight amount so as always to keep it just in the proper position. As the speed varied from low values to maximum value the pointer moved a distance of about 2 in. so that the change in bore was quite perceptible.

21 Owing to the fact that the increase of bore varies with the square of the speed there was no perceptible change below about 750 r.p.m., and this was the lowest speed used. The highest speed was 3900 r.p.m.

CALIBRATING SLEEVE TEST

22 In order to find the relation between the readings of the pointer and the actual values of increase of bore, and also to demonstrate that the apparatus would give correct measurements at high speed, a calibrating sleeve was made. This was a thin sleeve bored out at each end so as to be a sliding fit on the shaft and with two different bore diameters in the central portion, either of which could be brought over the pistons of the measuring device. Readings of the pointer were taken at each of these bores, stationary and rotating, at various speeds. The stationary readings gave the motion of the pointer for the known change of bore calibrating the

entire apparatus. The readings while rotating showed that the instrument gave correct readings at all speeds.

TEST WITH ROTATING WHEEL

23 Next the rotating wheel was pressed on the shaft. A number of readings of the diameter of shaft and bore of wheel showed that the average amount of force fit was 0.0017 in. After this, a pair of outside calipers was set to fit the shaft at various points and a pair of inside ones set to fit the wheel bore. The difference between the two calipers was then found by means of a thickness gage. The mean of these observations gives 0.0011 in. as the amount of force fit. The mean of these two values 0.0014 in. is considered to be the actual amount of force fit. It was necessary to know the amount of this force fit in order to make comparisons between the computed and observed amounts of increase of bore of the rotating wheel, since the wheel at zero speed is expanded somewhat.

TABLE 1 READINGS OF POINTER IN TESTS WITH ROTATING WHEEL, CONDUCTED MARCH 11, 1911

R.P.M.	Test No. 9		Test No. 10		Test No. 11		Test No. 12		Test No. 13	
	Up	Down	Up	Down	Up	Down	Up	Down	Up	Down
0	2.61	2.62	2.62	2.63	2.62	2.63	2.63	2.62	2.61	2.62
750	2.61	2.54	2.63	2.59	2.63	2.59	2.64	2.62	....	....
1050	2.64	2.58	2.70	2.68	2.68	2.67	2.66	2.70	....	....
1200	2.69	2.63	2.74	2.72	2.72	2.74	2.72	2.75	....	....
1500	2.72	2.70	2.80	2.82	2.82	2.80	2.84	2.90	....	....
1800	2.78	2.79	2.90	2.96	2.92	2.90	2.96	3.00	....	....
2100	2.84	2.89	3.03	3.10	3.02	3.10	3.11	3.20	....	....
2400	2.99	3.14	3.18	3.25	3.18	3.24	3.26	3.40	....	....
2700	3.16	3.30	3.32	3.44	3.32	3.48	3.48	3.56	....	....
3000	3.30	3.45	3.54	3.63	3.54	3.63	3.70	3.75	....	....
3150	3.50	3.64	3.70	3.78	3.75	3.85	3.85	3.85	....	....
3300	3.65	3.70	3.85	3.94	3.85	3.98	4.00	4.00	....	....
3450	3.80	3.80	4.01	4.08	4.00	4.08	4.10	4.08	....	....
3600	4.00	4.00	4.13	4.20	4.15	4.18	4.24	4.20	....	....
3750	4.12	4.12	4.24	4.28	4.30	4.29	4.34	4.33	....	....
3900	4.30	4.30	4.34	4.34	4.38	4.38	4.38	4.38	4.38	4.38

24 Fig. 2 is a diagram of the arrangement of the apparatus when the wheel was on the shaft. Readings of the pointer were taken at various speeds as shown by Table 1 and the results plotted in Fig. 3. The pointer readings, as well as the corresponding values of actual increase in bore as shown by the previous calibrations, are given

on the left. Owing to the fact that the wheel was forced on the shaft, the measured diameter of the wheel at zero speed is greater than the actual free diameter of the wheel. By methods given in Pars. 29-39, it was computed that the wheel was expanded by the force fit one-half of the difference in free diameters, the remainder being compression of the shaft. Hence the base line for increase in bore of the wheel from the free condition is the lower line of Fig. 3, which is 0.0007 in. below the base line for increase in bore from the

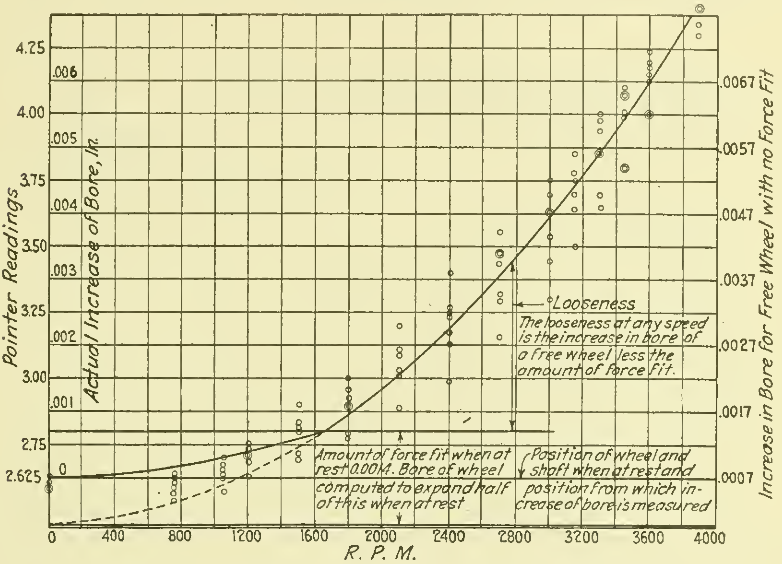

Fig. 3 Increase in Bore of a Rotating Wheel

Points from Test with Rotating Wheel (Table 1); Curve drawn from Computations.

actual initial value when on the shaft. The computed curve for the increase in bore for the free wheel at the various speeds is drawn in Fig. 3 beginning with the lower base line. At that speed where the expansion of the free wheel is equal to the amount of the force fit, 0.0014 in., the shaft is just released from its compression and wheel and shaft are just of the same diameter. For speeds beyond this point the wheel expands exactly as if there was no initial force fit. The wheel is loose on the shaft by whatever increase in bore occurs beyond this point. The observed points are seen to fit very closely the computed curve. For all speeds below that at which the wheel and shaft just touch there is compression of the shaft and expansion of the wheel by the force fit, the amount being successively less as the speed increases. The effective force fit at any speed is the difference between the force fit at rest, and the expansion of the free wheel at the given speed. The effective force fit is, therefore, the difference between the dotted curve and the upper horizontal line. As shown later, when there are different amounts of force fit with a given wheel and shaft, the expansion of the wheel is always a constant fraction of the amount of force fit. This is one-half in the present case. Hence the curve showing the increase in bore of the wheel, as the force fit is gradually relieved, is drawn halfway between the dotted curve and the upper horizontal line.

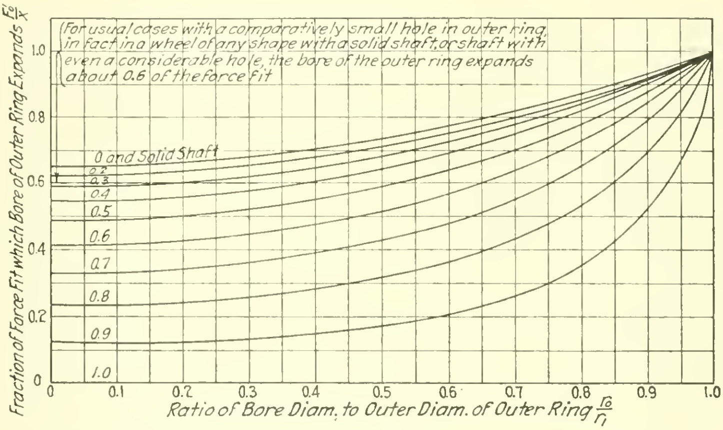

Fig. 4 Ratio of Expansion of Bore to Force Fit
Figures on Curves denote Ratio of Diameter of Hole in Inner Ring to Diameter of Bore of Outer Ring $\frac{r_2}{r_0}$

25 In the present case the amount of the force fit is comparatively small. It was originally intended to repeat the experiments with successively increasing amounts of force fit. The measuring apparatus was arranged so that it could be torn down and reassembled without change in the absolute initial value of the readings. This was done with the idea of actually measuring the change in the initial bore of the wheel when put on the shaft with different amounts of force fit, giving curves in Fig. 3 which would start with successively greater amounts of force fit and meet the curve of a free wheel higher up. However, the results of the first experiment agreed so closely with the theory and everything was

so obvious after study of the subject that no further experiments were made.

DESIGN OF WHEELS SUBJECT TO INCREASE OF BORE BY CENTRIFUGAL STRESSES

26 The method used to prevent difficulty due to loose wheels is to force the wheel on the shaft with a force fit, or difference in diameters between wheel bore and shaft, greater than the computed amount of increase of bore at maximum speed. In some cases this has resulted in large amounts of force fit requiring very heavy pressures for forcing. Special material is necessary for the bushings between the wheel and the shaft. However, all of these points have been successfully met and ill effects due to loosening of the wheels wholly eliminated.

27 In some cases the wheel hubs do not fit along their entire length but only in places. The radial pressure when the fit only occurs for a short distance is obviously increased by the ratio of the total length of hub to the actual length of the fit. The numerical values of this radial pressure on the bushings, computed by methods discussed in Par. 33, come to very high figures in many cases. Of course, some relief is offered by the fact that this pressure is pure compression and that the metal of the bushings is constrained on nearly all sides so that the resistance to flow is considerable. Under such circumstances much higher values of compression stress can be used than when there is no resistance to the deformation which the compression tends to cause. There is reason to believe that in the ideal case of a piece subject to pure compression in every direction, there would be no failure even with an infinitely large stress.

28 It is to be remarked that when the force fit is just sufficient to prevent loosening at full speed and the shaft is solid, the effective stress in the hub due to the force fit when at rest, is exactly equal to the stress at full speed.

COMPUTATION OF FORCE AND SHRINK FITS WITH STEEL

29 A complete treatment of the subject based on the maximum shear law previously mentioned is given for the first time so far as known. The general nature of ring stresses is in Pars. 3-8. The mathematical derivation of the formulae is given in the Appendix.

30 It turns out that a wheel hub is affected only for a very short distance outward so that the nature of the outer portion has little or no influence. The condition near the hub in a flat disk with an

outer diameter only ten times the diameter of the bore is practically the same as in a disk with an infinitely large outer diameter. We therefore consider only the case of a ring forced upon another ring of equal axial length. We estimate the outer diameter of the outer ring so as to make it equivalent so far as expansion is concerned to a given wheel of any shape. For instance, for a solid wheel comparatively thin at the outer portion, we would estimate the equivalent flat disk to have one-half or one-third the diameter of the wheel.

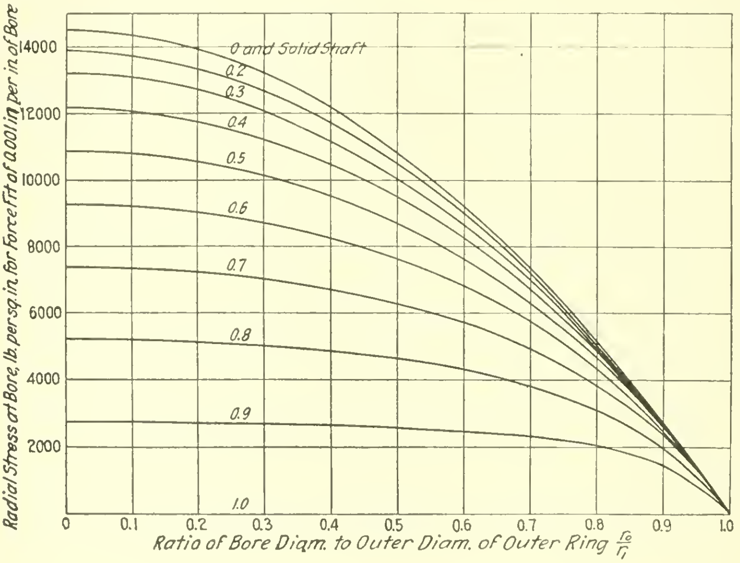

Fig. 5 Radial Stress at Bore S_{r0}. Force fit of 0.001 In. per In. of Bore Stress is directly proportional to Force Fit per Inch of Bore. Figures on Curves denote Ratio of Diameter of Hole in Inner Ring to Diameter of Bore of Outer Ring $\frac{r_2}{r_0}$.

For a wheel with spokes, we have little more than the hub only. We also estimate an equivalent inner ring if we do not have a solid or hollow shaft. Owing to the comparatively slight influence of the parts of the rings furthest from the fit, a considerable error in estimation of the diameters of the equivalents will cause comparatively small error in the results.

31 Figs. 4, 5 and 6 give amount of expansion and stresses in an outer ring or hub pressed or shrunk on inner ring or shaft.

32 The difference in the free diameters is called the force fit, which we will denote by X. When the hub is in place on the shaft and there is no rotation so as to loosen, it is expanded by a certain amount F'_o. The shaft is also compressed by a considerable amount. The sum of the increase of hub bore, F'_o, and the compression of the shaft is evidently the amount of the force fit. Fig. 4 gives values of $\frac{F'_o}{X}$ the fraction of the force fit which the hub expands.

33 Fig. 5 gives the radial stress at the bore of the hub, which is also the stress at the outside of the shaft, as well as the pressure per square inch between when there is actual contact for the full length of hub.

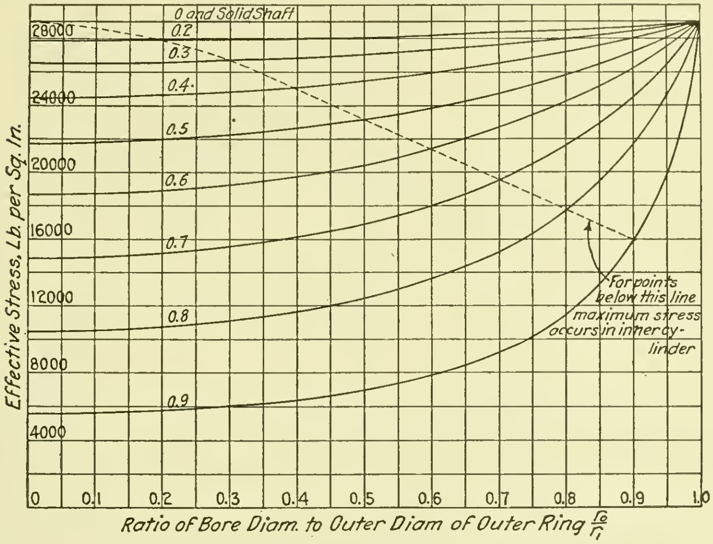

Fig. 6 Effective Tensile Stress at Bore of Outer Ring. Force Fit of 0.001 In. per In. of Bore

Stress is directly proportional to Force Fit per Inch of Bore. Figures on Curve denote Ratio of Inner Diameter of Inner Ring to Bore Diameter $\frac{r_2}{r_0}$.

34 Fig. 6 gives the maximum effective stress at the bore of the hub. This is the greatest stress in the system unless the shaft is hollow and relatively thin. Figs. 5 and 6 are for a force fit of one mil per inch of bore. Stresses for other values are in proportion.

35 The abscissae of these figures give the ratio of the bore to the outer diameter of the hub. The upper curve in each of these

three figures is for a solid shaft or one with a small hole and the other curves are for rings with various sizes of holes. When the abscissa is zero the outer diameter of the hub is infinitely large as compared with the bore. The curves are nearly horizontal for a considerable distance in this region and the results for an outer diameter ten times the diameter of the bore, that is, a ratio of 0.1, are almost the same as for an infinite outer diameter. The values for an outer diameter five or three times the diameter of the bore change but little.

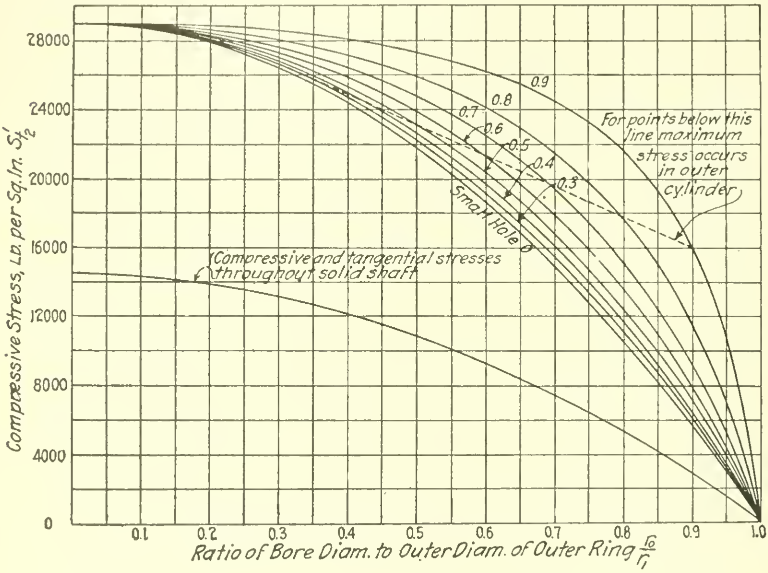

Fig. 7 Compressive Tangential Stress S'_{t2} at Inner Diameter of Inner Ring. Force Fit of 0.001 In. per In. of Bore

Stress is directly proportional to Force Fit per Inch of Bore. Figures on Curves denote Ratio of Inner Diameter of Inner Ring to Bore Diameter $\frac{r_2}{r_0}$. Values given are the Maximum Effective Stresses in Inner Ring.

36 Similarly, as will be seen by noting the curves for the various values of the ratio of the diameter of the hole in the shaft to the shaft diameter, the hole has but little influence unless it is comparatively large. The usual case of a shaft with a comparatively small hole is practically equivalent to a solid shaft. As will be seen by a study of the curves, the case where the diameter of the hub is seven times the diameter of the bore, giving a ratio of 0.14, and

where the hole in the shaft is 0.3 of the shaft diameter, gives very nearly the results of any usual case. Hence, for usual cases the hub bore expands about 0.6 of the amount of the force fit and for a force fit of one mil per inch of bore the radial stress is about 13,000 lb. per sq. in. and the effective tangential stress is about 27,000 lb. per sq. in.

37 For a solid shaft, however, the maximum effective stress is 29,000 lb. per sq. in. for a force fit of 1 mil per in. of bore, regardless of the character of the wheel or hub. The stress for other amounts of force fit is in proportion.

38 Force fits of 1 mil per in. are therefore about as large as will give no permanent set of the hub for steel and a solid shaft. Force fits up to about 1½ mils are frequently used. These give some set of the fibers very near the bore. This set is almost entirely eliminated by the elastic force of the outer fibers if the hub is ever removed from the shaft, however.

39 Fig. 7 gives the maximum stress (at the inside) in the shaft or inner ring. This is usually less, but for a relatively thin ring may be greater, than the stress in the hub. A solid shaft has a stress one-half as great as one with a small hole.

40 A rational expression for the force in pounds to press the hub on the shaft is

$$0.12\ d_o l S_{ro}$$

where l is the length, d_o the diameter of bore of the hub in inches and S_{ro} the radial stress at the bore found from Fig. 5. The coefficient of 0.12 was obtained from actual tests and gives a coefficient of friction of 0.038.

41 For a force fit of one mil per inch of bore the forcing pressure in our usual case is about 1560 lb. per in. of bore and per in. of hub length.

42 The forcing pressure of course varies widely on account of the influence of surfaces and lubricant. The formula gives an average value.

APPENDIX

DERIVATION OF FORMULAE FOR FORCE AND SHRINK FITS[1]

NOTATION

S_r = Radial stress, lb. per sq. in.

S_t = Tangential stress, lb. per sq. in.

r = Radius, inches

E = Modulus of elasticity taken 29,000,000 lb. per sq. in.

V = Poisson's ratio, taken 0.3

X = Amount of force fit, or difference in diameters, inches

F = Increase of diameter, inches

40 We take the case of two steel rings, the outer one being forced on the inner one. A subscript after any symbol denotes the value for a particular radius: 1 denotes the outer diameter of the outer ring or hub; 0 denotes the common diameter where the force fit occurs and 2 denotes the bore of the inner ring. We will also use an index ′ to denote values for the outer ring in general and indexes ″ to denote the inner ring.

41 The increase in radius of an elementary ring is $\frac{F}{2}$ and the tangential strain is

$$\frac{F}{2r} = \frac{S_t - VS_r}{E}$$

The radial strain is

$$\frac{1}{2}\frac{dF}{dr} = \frac{S_r - VS_t}{E}$$

and by using the preceding value of F we obtain a fundamental equation applying to any circular body.

$$V\left(S_r + r\frac{dS_r}{dr} - S_t\right) = S_t + r\frac{dS_t}{dr} - S_r \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

42 The statement that the radial forces on an elementary segment in a stationary cylinder are balanced is

$$S_r dr\,d\theta + r\,d\theta\,d\,S_r - 2\,S_t dr \sin d\frac{\theta}{2} = 0$$

This reduces to a second fundamental differential equation

$$S_r + r\frac{dS_r}{dr} - S_t = 0 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [5]$$

This enables reduction of the first fundamental equation to the form

---

[1] This section was written with the assistance of Mr. C. A. Schellens of the General Electric Company, Lynn, Mass.

$$S_t+r\frac{dS_t}{dr}-S_r=0 \qquad [6]$$

We integrate these simultaneous differential equations by adding and integrating, giving

$$S_r+S_t=2A$$

We add this to the second fundamental equation which separates the variables. By multiplying by r we obtain the exact differential of r^2S_r so that we obtain the integral directly. From the equation above we obtain the other integral, giving the equations

$$S_r'=A'-\frac{B'}{r^2}$$

$$S_t'=A'+\frac{B'}{r^2}$$

$$S_r''=A''-\frac{B''}{r^2}$$

$$S_t''=A''+\frac{B''}{r^2}$$

43 A' B' A'' B'' are four constants to be determined by the boundary conditions. Two of these are that the radial stresses at the inner and outer radii r_1 and r_2 are zero for the case of a hole in the inner ring. That is

$$S'_{r1}=0$$

$$S''_{r2}=0$$

Substituting these values in the above equations

$$0=A'-\frac{B'}{r_1^2}$$

$$0=A''-\frac{B''}{r_2^2}$$

44 This enables us to eliminate two constants giving a new set of fundamental equations

$$S_r'=B'\left(\frac{1}{r_1^2}-\frac{1}{r^2}\right)$$

$$S_t'=B'\left(\frac{1}{r_1^2}+\frac{1}{r^2}\right)$$

$$S_r''=B''\left(\frac{1}{r_2^2}-\frac{1}{r^2}\right)$$

$$S_t''=B''\left(\frac{1}{r_2^2}+\frac{1}{r^2}\right)$$

45 The third boundary condition is that the radial stresses at the common surfaces are equal, that is

$$S'_{r0}=S''_{r0}$$

giving

$$B'\left(\frac{1}{r_1^2}-\frac{1}{r_0^2}\right)=B''\left(\frac{1}{r_2^2}-\frac{1}{r_0^2}\right)$$

46 The fourth condition is that the algebraic difference between the deformations at the common radius is equal to the force fit. That is

$$F_0'-F_0''=X$$

47 In general the radial deformation at any radius is given by

$$\frac{F}{2}=\frac{r}{E}\left(S_t-VS_r\right)$$

Outward deformation and tensile stress are counted positive.

48 Then we have, after noting that $S'_{r0}=S''_{r0}$

$$\frac{2r_0}{E}\left\{S'_{t0}-S''_{t0}\right\}=X$$

Substituting the stresses from the last set of equations and substituting for B'' from the equation given by the third boundary condition, this becomes

$$B'\left\{\frac{1}{r_1^2}+\frac{1}{r_0^2}-\frac{\left(\frac{1}{r_2^2}+\frac{1}{r_0^2}\right)\left(\frac{1}{r_1^2}-\frac{1}{r_0^2}\right)}{\left(\frac{1}{r_2^2}-\frac{1}{r_0^2}\right)}\right\}=\frac{EX}{2r_0}$$

Hence

$$B'=\frac{EX}{4r_0}\,\frac{r_1^2\left(1-\frac{r_2^2}{r_0^2}\right)}{\frac{r_1^2}{r_0^2}-\frac{r_2^2}{r_0^2}}\ldots\ldots\ldots\ldots[7]$$

This gives a value of B' which we can substitute in the fundamental equations. The value of the radial stress at the common radius then comes out

$$S'_{r0}=-E\frac{X}{4r_0}\,\frac{\left(\frac{r_1^2}{r_0^2}-1\right)\left(1-\frac{r_2^2}{r_0^2}\right)}{\frac{r_1^2}{r_0^2}-\frac{r_2^2}{r_0^2}}\ldots\ldots\ldots\ldots[8]$$

This is a compressive stress. The stress varies in the outer ring from the above value at the bore to zero at the outside.

49 Fig. 5 gives values computed from this equation. Values of radial stress S_{r0} are ordinates, for a value of $\frac{X}{2r_0}$, the force fit per inch of bore, of one mil per inch. Values of $\frac{r_0}{r_1}$ are abscissae. Each of the curves is for a given value of $\frac{r_2}{r_0}$.

50 The fraction of the force fit which the outer ring expands is given by

$$\begin{aligned}\frac{F_0'}{X}&=\frac{2r_0}{EX}\left(S'_{t0}-VS'_{r0}\right)\\&=\frac{2r_0B'}{EX}\left\{\frac{1}{r_1^2}+\frac{1}{r_0^2}+V\left(\frac{1}{r_0^2}-\frac{1}{r_1^2}\right)\right\}\\&=\frac{1}{2}\,\frac{\left\{1+\frac{r_1^2}{r_0^2}+V\left(\frac{r_1^2}{r_0^2}-1\right)\right\}\left(1-\frac{r_2^2}{r_0^2}\right)}{\frac{r_1^2}{r_0^2}-\frac{r_2^2}{r_0^2}}\ldots\ldots\ldots\ldots[9]\end{aligned}$$

51 Fig. 4 gives values computed from this equation. The ordinates are values of $\frac{F'_0}{X}$, the fraction of force fit which the outer ring expands. Values of $\frac{r_0}{r_1}$

are abscissae. Each of the curves is for a given value of $\frac{r_2}{r_0}$.

52 If we substitute the value of B' in the last set of fundamental equations and obtain the value of S'_t we find that there is a tangential tensile stress in the outer cylinder with greatest value at the bore. Hence, since the radial stress at this point is compressive, we have as the greatest value of the effective stress in the outer cylinder, at the bore

$$S'_{t0} - S'_{r0} = E\frac{X}{2r_0}\,\frac{1-\frac{r_2^2}{r_0^2}}{1-\frac{r_2^2\,r_0^2}{r_0^2\,r_1^2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [10]$$

53 Fig. 6 gives values computed from this equation. For a small hole in the inner cylinder, and also, as will be shown later, for a solid shaft, this maximum stress in the outer cylinder, at its bore, becomes

$$E\frac{X}{2r_0}$$

This value of the maximum effective stress in a hub on a solid shaft, obtained in the correct way by adding the tangential stress due to hub extension and radial stress due to shaft compression, turns out to give the same final results as the incorrect method of finding the tangential stress in the hub with an incompressible shaft.

54 From the relation between B' and B'' and the value of B' we can substitute in the last set of fundamental equations and obtain the stresses in the inner ring. Both stresses are compression throughout so that the greatest gives the maximum effective stress direct.

55 The radial compressive stress increases from zero at the inner diameter to a maximum at the outer diameter of the inner ring, where it has the value already given for the bore of the outer cylinder and shown in Fig. 5. The tangential compression stress increases from the outer diameter to the diameter of the central hole where it is the maximum stress in the inner cylinder and is

$$S''_{t2} = -E\frac{X}{2r_0}\,\frac{1-\frac{r_0^2}{r_1^2}}{1-\frac{r_2^2\,r_0^2}{r_0^2\,r_1^2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [11]$$

Values of the expression are plotted in Fig. 7. It is to be noted that in case the ratios of radii of both rings are alike, the above expression for maximum stress in the inner ring coincides with the previous expression for maximum effective stress in the outer ring. It can also be seen that the greater of the two maximum stresses is in that ring which has the greatest ratio of inner to outer diameter, i. e., which is relatively thinnest.

56 The preceding work has been for the case where there was a hole in the inner cylinder. For a solid shaft, the inner boundary condition in the inner cylinder becomes

$$S''_{t2} = S''_{r2}$$

instead of

$$S''_{r2} = 0$$

the other conditions remaining the same. From the first set of fundamental equations and the above relation

$$A''r_2^2 - B'' = A''\,r_2^2 + B''$$

If we now put $r_2 = 0$ we have

$$B'' = 0 \text{ and } S''_r = S''_t = A''$$

That is, the radial and tangential stresses are equal and constant throughout a solid shaft. This is a well-known property of any solid with uniform normal stress over the entire surface.

57 By inserting the remaining boundary condition we obtain final values for the stresses in the outer cylinder exactly the same as the previous values for a hollow shaft when the radius of the inner hole r_2 is made equal to zero giving an infinitely small hole. The value of the uniform stress throughout the shaft is

$$S''_r = S''_t = -\frac{E}{2}\frac{X}{2r_0}\left(1 - \frac{r_0^2}{r_1^2}\right) \ldots\ldots\ldots\ldots\ldots\ldots [12]$$

58 Values of this expression are given by the lower curve of Fig. 7. It is exactly one-half the previous value of the stress at the center of a ring when the radius of the inner hole r_2 is made equal to zero, giving an infinitely small hole.

INVESTIGATION OF EFFICIENCY OF WORM GEARING FOR AUTOMOBILE TRANSMISSION

By Wm. H. Kenerson

ABSTRACT OF THE PAPER

The paper describes a series of tests made to determine the efficiency of three types of worm gearing intended for use in automobile transmission, and at the same time a series of tests to determine the effect of continuous running. Data regarding the shape and construction of the gears are given and the results tabulated and also expressed in graphic form. The results indicate a high efficiency and large carrying capacity.

INVESTIGATION OF EFFICIENCY OF WORM GEARING FOR AUTOMOBILE TRANSMISSION

By Wm. H. Kenerson, Providence, R. I.

Member of the Society

This investigation was made at the plant of the Brown & Sharpe Manufacturing Company for the purpose of determining the efficiency of three types of worm gearing for use in an automobile transmission system and the heating effect due to continuous running. The power for these tests was obtained from a 50-h.p. induction motor, running approximately 870 r.p.m. at full load.

2 Between the motor and the worm gear case was placed an automobile transmission gear case to enable tests to be made at two lower speeds. Between this and the worm gear case was placed a transmission dynamometer designed by the author and described in a paper presented before this Society and published in Vol. 31 of Transactions.[1] An Alden brake was used to absorb and measure the power transmitted by the gears under test. The arrangement of the apparatus is shown in Fig. 1 which shows the transmission dynamometer set up for comparison with the Alden brake, and in Fig. 2 which shows the whole apparatus mounted ready for testing the worm gear efficiency. Referring especially to Fig. 2, *A* is the motor, *B* the automobile transmission gear case, *C* the transmission dynamometer, *D* the case containing the worm gear under test, *E* the Alden brake, and *F* the platform scale which measured the load on the Alden brake.

3 The apparatus was so arranged that as the load was imposed the weight on the platform scale was removed and all vibration of the scale beam was eliminated by interposing springs *S* between the blocks *FF* which sustained the weight on the

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[1] Wm. H. Kenerson, A New Transmission Dynamometer, p. 171.

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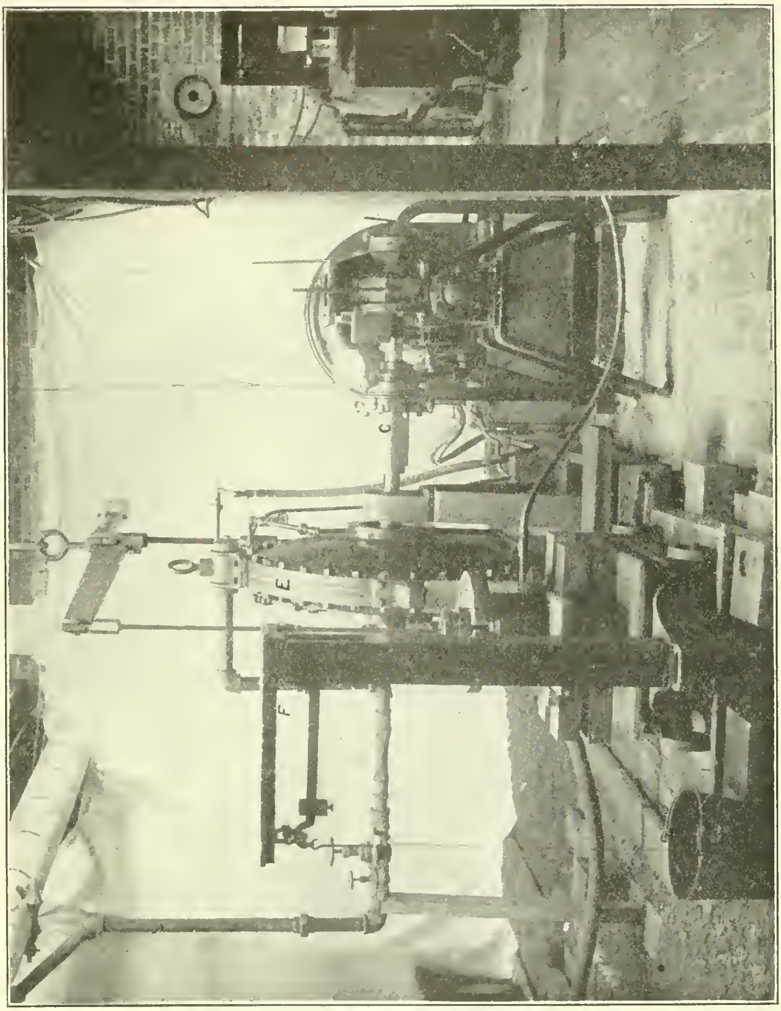


Fig. 1 Apparatus Arranged for Calibration

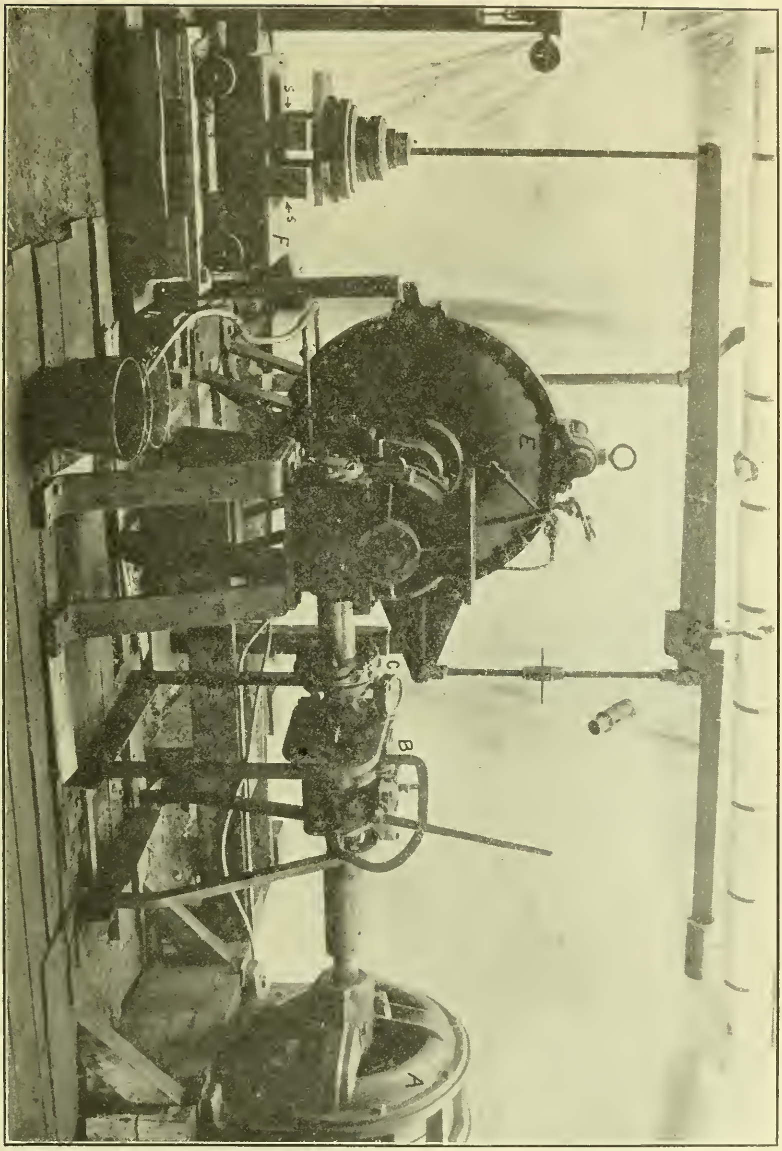


FIG. 2 APPARATUS FOR TESTING WORM GEARS

scale, and also by suspending by a wire from the weight at the end of the scale beam a plate which dipped into a pail of oil, thus acting as an efficient dashpot. It was found possible by careful manipulation to maintain a steady and easily read load on both the transmission and absorption instruments.

4 The transmission dynamometer and brake were first compared by arranging them as shown in Fig. 1, in which position

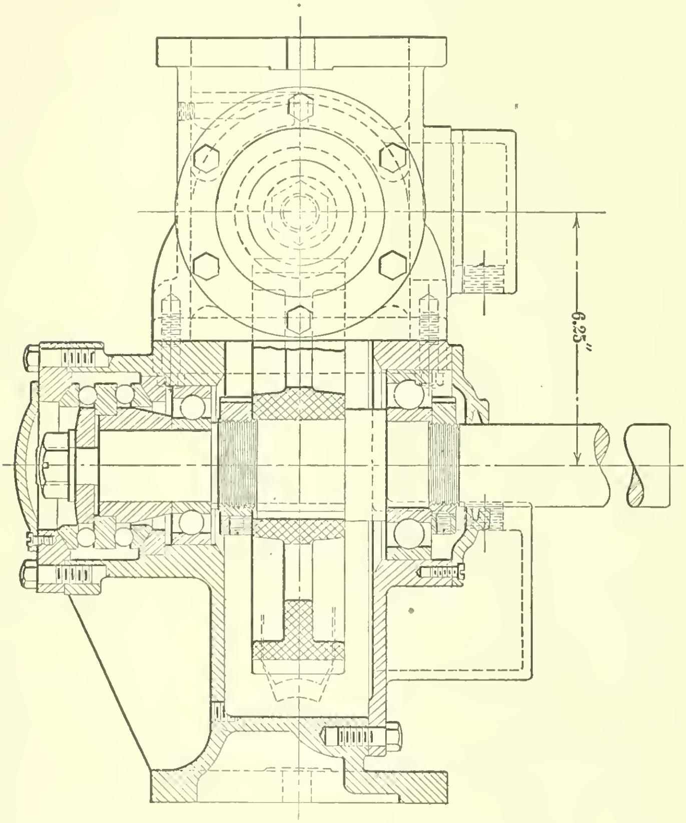


Fig. 3 Section through Gear Case

runs were made at various loads, and the torques corresponding to horsepower per 100 r.p.m. marked on the dial of the transmission instrument corresponding to similar loads as indicated on the brake. It is evident that by this method of comparison the two instruments must check each other exactly.

5 A thermometer placed in the oil well at the back of the

worm gear case *D* indicated the temperature of the oil, and another thermometer placed on the wall near the apparatus indicated the room temperature.

6 The oil employed to lubricate the gears was one intended for use with superheated steam, having a specific gravity of 26 Baumé, flash point 625 fahr., and viscosity at 212 deg. 260 to 265. The case contained about 5 quarts of the oil.

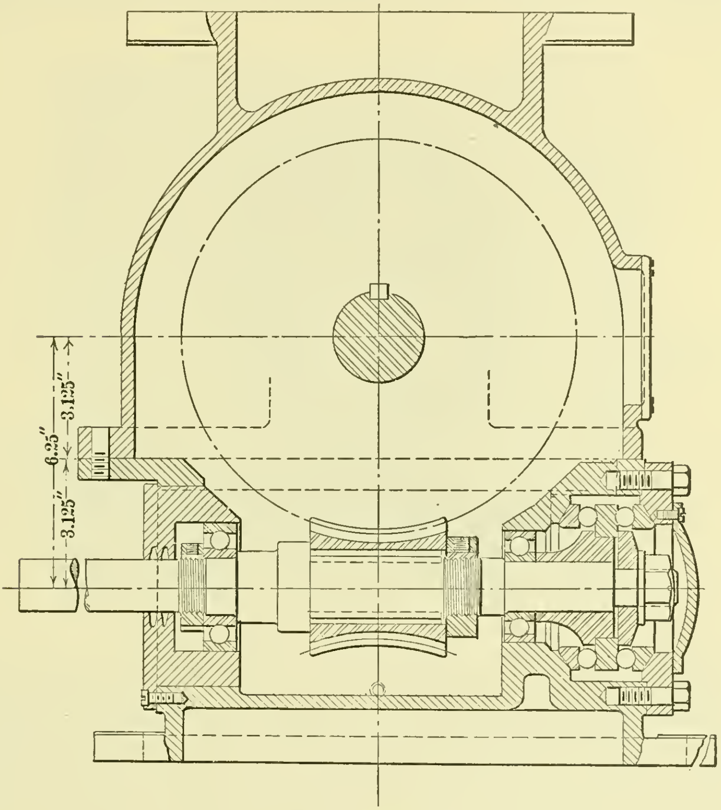


Fig. 4 Section through Gear Case

7 In all the trials the worm was located underneath the gear. Figs. 3 and 4 show sections through the gear case. As indicated, both shafts are mounted on ball bearings and end-thrust ball bearings take care of the thrust on the worm and wormwheel. All the worms were made of machinery steel, case-hardened, and the wormwheels of phosphor bronze.

8 The first worm and wheel tested are shown in Fig. 5, and are similar to those used for driving the spindle on the Brown & Sharpe automatic spur gear machines. This pair of gears is an unusual case of the worm and wormwheel. The smaller gear is hobbed with a hob of the size of the larger gear, thus making possible adjustment of the larger gear, which would not otherwise be the case. While in appearance this gear resembles a Hindley worm, it is not such in reality.

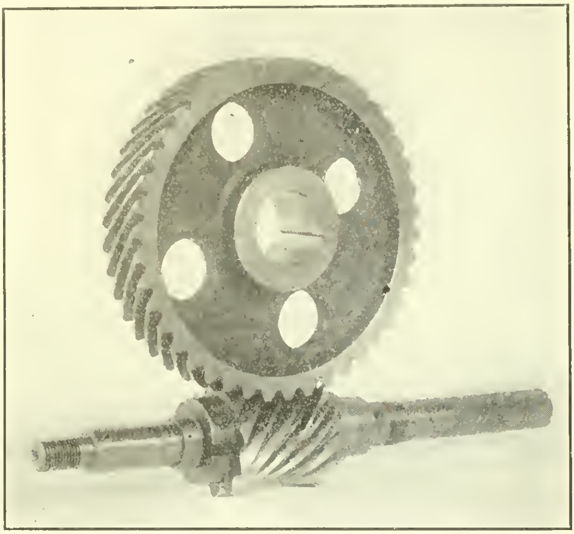


Fig. 5 First Worm and Wheel Tested

9 Although not strictly in accordance with usage, the smaller gear will be called the worm and the larger the wormwheel in the following description. Figs. 5 and 6 give the dimensions and a good general idea of this pair. The wormwheel was cut with the cutter shown, and the shape of the teeth on a section through the worm and wormwheel, parallel to the axis of the worm, is also shown.

10 All necessary information pertaining to pairs No. 2 and 3 is given in Figs. 7, 8 and 9. The difference between gears 2 and 3 is principally one of shape of the worm threads. This difference is clearly brought out in the sections of the worm and wheel shown in Figs. 10 and 11.

11 The shape of the teeth on the worm in Fig. 10 was produced with a cutter, the included angle of which was 29 deg., and the depth of tooth 0.570 in. This depth was based on the axial pitch, when the usual method on multiple worms is to base the depth on the normal pitch. The object in using this cutter was to obtain as many teeth as possible in contact at one time, and also a shape that could be ground with a straight-sided emery wheel.

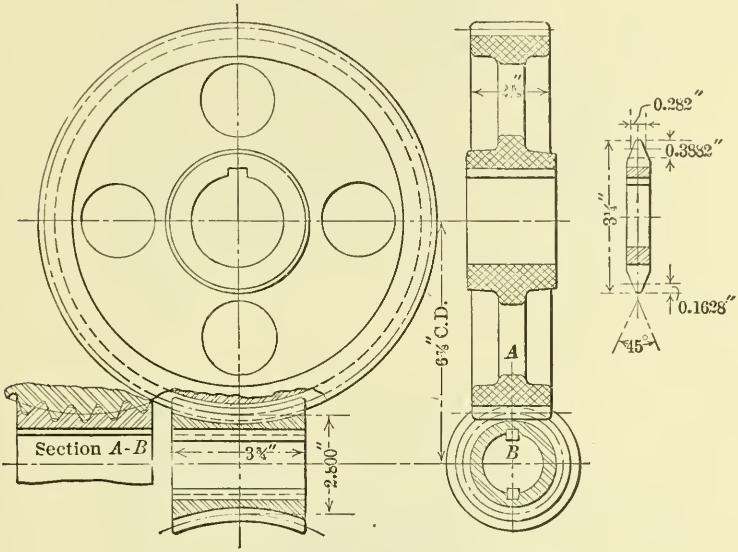


Fig. 6 No. 1 Worm and Wormwheel

Wormwheel, Phosphor Bronze. No. of teeth, 43 L. H.; Pitch diameter, 10.95; Outside diameter, 11.28; Circular pitch, 0.800; Angle of teeth with axis, 45 deg.; Normal circular pitch, 0.5657; Pitch of cutter, 5.553; Addendum, 0.1628 (not standard); Thickness of tooth, 0.282; Whole depth, 0.3882; Included angle of cutter, 45 deg.

12 The teeth of the worm in Fig. 11 were cut with a cutter shaped as shown, which is an arbitrary shape made to produce at one instant the greatest effective breadth possible, as shown in the figure.

13 In conducting the trials the load was maintained at the desired point by one observer who adjusted the brake. Readings were then taken on the transmission instrument by two independent observers. The speed of the motor was observed in each

case and from this, knowing the gear ratio, all the other speeds were easily computed. Temperatures were taken immediately following each series of observations.

14 For purposes of comparison a series of trials was run on a pair of bevel gears, the principal dimensions of which are shown in Fig. 12. Table 1 is a record of one set of observations typical of the series. Table 2 is a summary of the results of all the trials, and the curves of Fig. 13 show in graphic form the average efficiency of the different gears at the various loads and speeds.

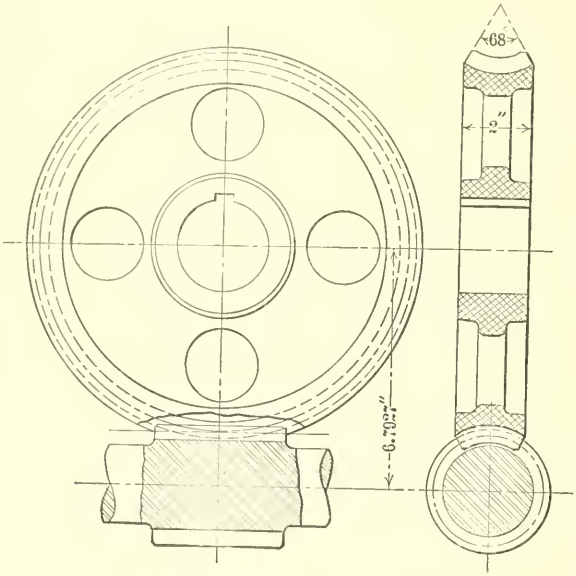


Fig. 7 Nos. 2 and 3 Worm and Wormwheel

Wormwheel, phosphor bronze; Number of teeth, 40; Pitch diameter, 10.5704; Throat diameter, 10.9964; Circular pitch, 0.8302; Angle of teeth with axis, 38° 16′ 5″; Normal circular pitch, 0.6518; Pitch of cutter, 4.8196; Addendum, 0.213; Thickness of tooth, 0.3568; Whole depth, 0.4586; Worm: Aurora Street, Case-Hardened; Number of teeth, 9; Pitch diameter, 3.015; Outside diameter, 3.441; Circular pitch, 1.0524; Angle of teeth with axis, 51° 43′ 55″; Thickness of tooth, 0.295; Lead, 7.4719; Ratio of Wheel to Worm, 40 to 9.

15 In conjunction with the efficiency trials a series of runs was made to determine the heating effect due to continuous running. In these trials, which were in effect endurance tests, a

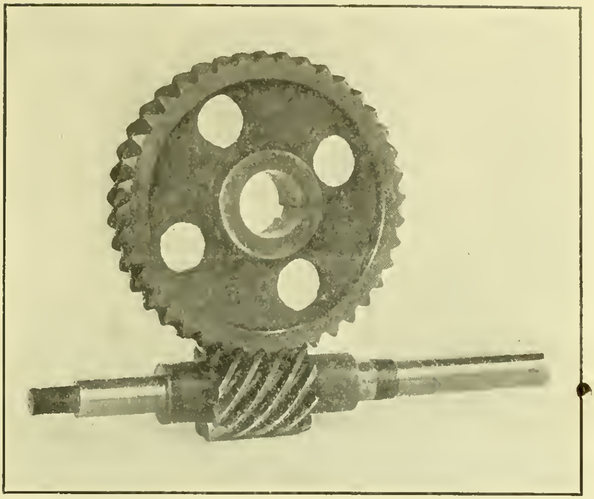


Fig. 8 No. 2 Worm and Wormwheel

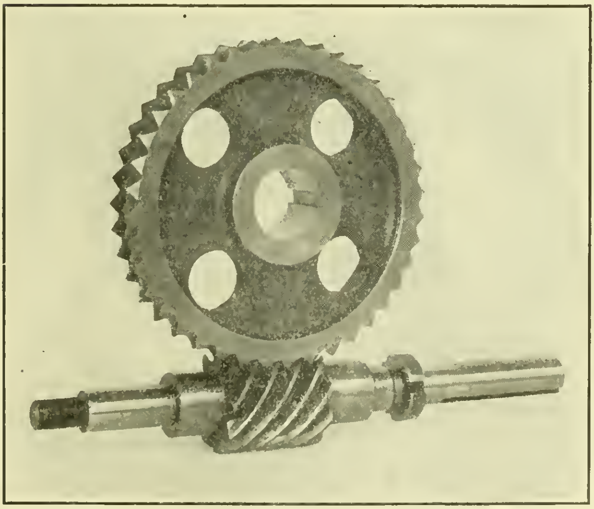


Fig. 9 No. 3 Worm and Wormwheel

TABLE 1 EFFICIENCY TEST OF NO. 2 WORM GEARS WHEN ON DIRECT SPEED

R.P.M. Motor	R.P.M. Worm	R.P.M. Worm Gear	Speed of Worm in Ft. per Min.	Initial Load	Reading on Scale	Load on Brake 315 Ft. Radius	H.P. per 100 R.P.M. Brake	Total Brake H.P.	Total Trans. Dyn. H.P.	Efficiency of Worm Gears	Transmission Dyn. Reading				Temp. at End of Test		Time		Duration
											1	2	3	Av.	Oil	Room	Start	Fin.	
875	875	196.9	690.7	104	55	49	24.5	48.24	49.26	97.9	5.6	5.65	5.65	5.63	268	65	10:14	10:20	6
873	873	196.4	689.1	104	60	44	22.0	43.22	44.35	97.5	5.1	5.05	5.1	5.08	273	65	10:20	10:25	5
880	880	198.0	694.6	104	65	39	19.5	38.61	40.92	94.4	4.7	4.65	4.6	4.65	276	66	10:25	10:30	5
880	880	198.0	694.6	104	70	34	17.0	33.66	36.26	92.8	4.15	4.1	4.1	4.12	275	66	10:30	10:33	3
883	883	198.7	697.0	104	75	29	14.5	28.81	31.17	92.4	3.55	3.55	3.5	3.53	273	66	10:33	10:37	4

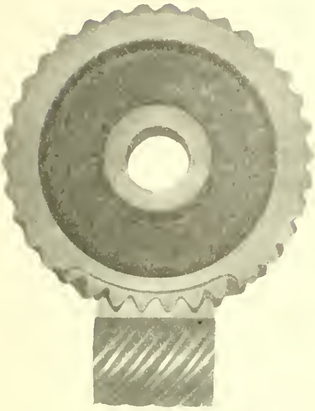
NUMBER OF GROUPS OF READINGS TAKEN IN TABLE 2

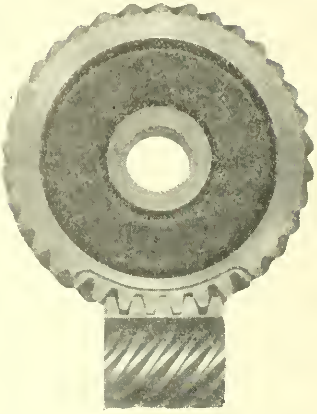
	Gear No. 1	Gear No. 2	Gear No. 3	Bevel Gear
First Speed....	4	5	5	3
Second Speed..	4	5	5	6
Direct........	3	5	5	5

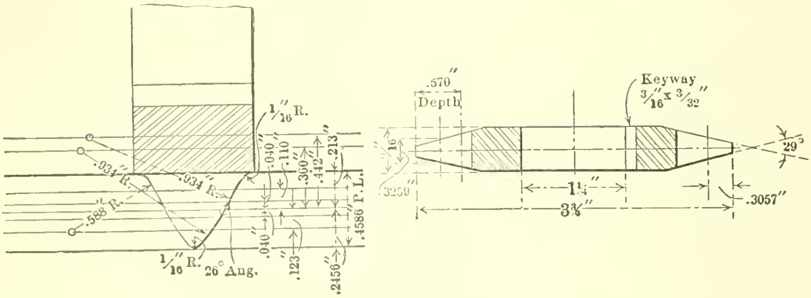
Total Separate Readings; 165

TABLE 2 WORMS AND WHEELS AND BEVEL GEARS, AVERAGE EFFICIENCIES TAKEN

	Gear No. 1							Gear No. 2							Gear No. 3							Bevel Gears						
	R.P.M.		H.P.		Temp.		Eff.	R.P.M.		H.P.		Temp.		Eff.	R.P.M.		H.P.		Temp.		Eff.	R.P.M.		H.P.		Temp.		Eff.
	Worm	Worm Wheel	Input	Output	Oil	Room		Worm	Worm Wheel	Input	Output	Oil	Room		Worm	Worm Wheel	Input	Output	Oil	Room		Pinion	Gear	Input	Output	Oil	Room	
First Speed	254.2	65.03	26.18	24.06	206	73	91.9	254.7	57.32	19.26	18.34	164	69	95.3	254.1	57.19	18.75	18.30	136	79	95.5	254.2	68.43	20.92	20.19		70	96.6
	254.6	65.13	24.75	22.47	217	73	90.8	254.7	57.32	17.99	16.91	169	69	94.1	254.8	57.33	16.54	15.48	144	79	93.1	254.5	68.51	17.50	16.78		70	95.9
	254.5	65.11	22.98	20.81	225	73	90.5	254.5	57.25	16.73	15.46	162	68	92.5	255.0	57.38	13.75	12.62	149	79	91.8	256.1	68.95	14.54	13.44		69	92.4
	255.1	65.25	21.17	19.25	228	73	90.9	254.3	57.22	15.08	14.02	142	69	93.1	255.4	57.47	10.80	9.77	150	79	90.5	256.5	69.05	10.95	10.01		70	91.4
	255.3	65.30	19.56	17.63	232	73	90.1	255.1	57.39	13.52	12.63	159	69	93.5	255.6	57.52	7.38	6.90	149	79	93.6							
	255.7	65.41	17.76	16.02	233	74	90.2	254.9	57.35	12.14	11.18	160	69	92.2														
	255.9	65.48	16.00	14.41	233	74	90.1	254.8	57.34	10.69	9.75	160	69	91.4														
	256.3	65.60	14.34	12.79	233	74	89.4																					
	256.5	65.62	12.37	11.15	231	74	90.2																					
	256.8	65.69	10.55	9.53	229	74	90.2																					
Second Speed	545.2	139.5	43.48	41.16	178	73	94.8	544.2	122.4	36.95	36.11	178	70	97.7	543.2	122.2	38.59	36.06	204	77	93.4	541.8	145.0	39.64	39.37		67	99.3
	546.9	139.9	40.54	37.78	207	70	93.0	545.0	122.6	34.61	33.10	184	70	95.6	543.8	122.4	35.36	33.04	215	77	92.4	544.1	146.4	33.43	32.22		67	96.4
	547.8	140.1	36.88	34.33	214	70	93.2	546.7	123.0	32.09	30.14	188	70	93.9	544.8	122.6	33.12	30.03	219	77	90.7	545.7	146.9	26.25	24.98		67	95.2
	548.8	140.4	33.47	30.89	217	70	92.3	548.4	123.4	28.90	27.15	187	70	93.9	545.8	122.8	29.82	27.02	221	77	90.6	548.5	147.6	18.75	17.72		67	94.5
	550.2	140.7	29.76	27.44	218	70	92.3	549.0	123.5	25.80	24.08	186	71	93.3	546.5	123.0	26.70	23.98	222	78	89.8							
	550.5	140.8	26.01	23.94	218	71	91.3	549.6	123.7	22.59	21.03	163	70	93.1	547.2	123.2	23.56	20.95	222	78	88.9							
	551.1	141.0	22.22	20.45	217	71	92.0	551.0	124.0	19.12	17.98	163	70	94.0	548.3	123.4	20.02	17.89	221	78	89.4							
Direct	859.5	219.8	66.43	64.86	178	69	97.6	873.5	196.5	49.35	48.14	265	66	97.5	866.0	194.8	49.95	47.72	215	77	95.5	872.0	234.7	52.05	51.63		70	99.2
	864.5	221.1	62.89	59.71	201	70	94.9	875.0	196.9	44.50	43.30	256	65	97.3	868.2	195.3	45.68	42.98	223	77	94.1	878.6	236.6	42.09	40.22		70	95.5
	871.0	222.8	59.23	54.60	214	70	92.2	878.0	197.5	40.40	38.50	260	66	95.2	870.8	195.9	41.20	38.20	228	77	92.7	884.4	238.2	30.40	28.58	132	79	94.0
	874.5	223.7	54.22	49.22	224	70	90.8	881.0	198.2	36.40	33.70	261	66	92.4	874.0	196.7	36.88	33.45	230	77	90.6							
	878.0	224.6	48.60	43.80	230	70	90.1	883.4	198.8	30.73	28.84	262	66	93.8	876.0	197.1	31.45	28.60	230	78	90.9							
	882.0	225.6	42.25	38.36	233	70	90.8								878.8	197.7	26.45	23.73	229	78	89.7							
	884.0	226.1	36.24	32.79	237	71	90.5																					







Figs. 10 and 11 Section of Nos. 2 and 3 Worm and Wormwheel

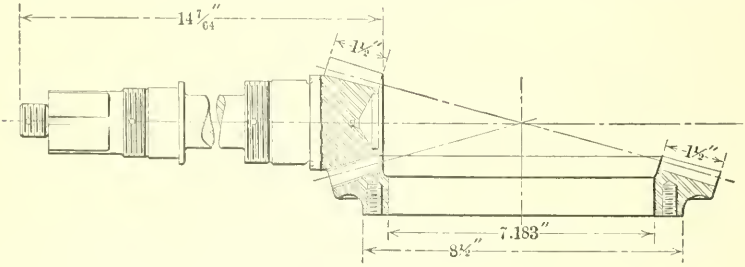


Fig. 12 Driving Gear and Pinion

Driving Pinion, 5 Per Cent Nickel Steel, Case-Hardened; Pitch, 5; Number of teeth, 14; Angle of edge, 15° 4′; Angle of face, 71° 5′; Outside diameter, 3.3359. Driving gear, 5 per cent nickel steel, case-hardened; Pitch, 5; Number of teeth, 52; Angle of edge, 74° 56′; Angle of edge, 13° 47′; Outside diameter, 10.4627.

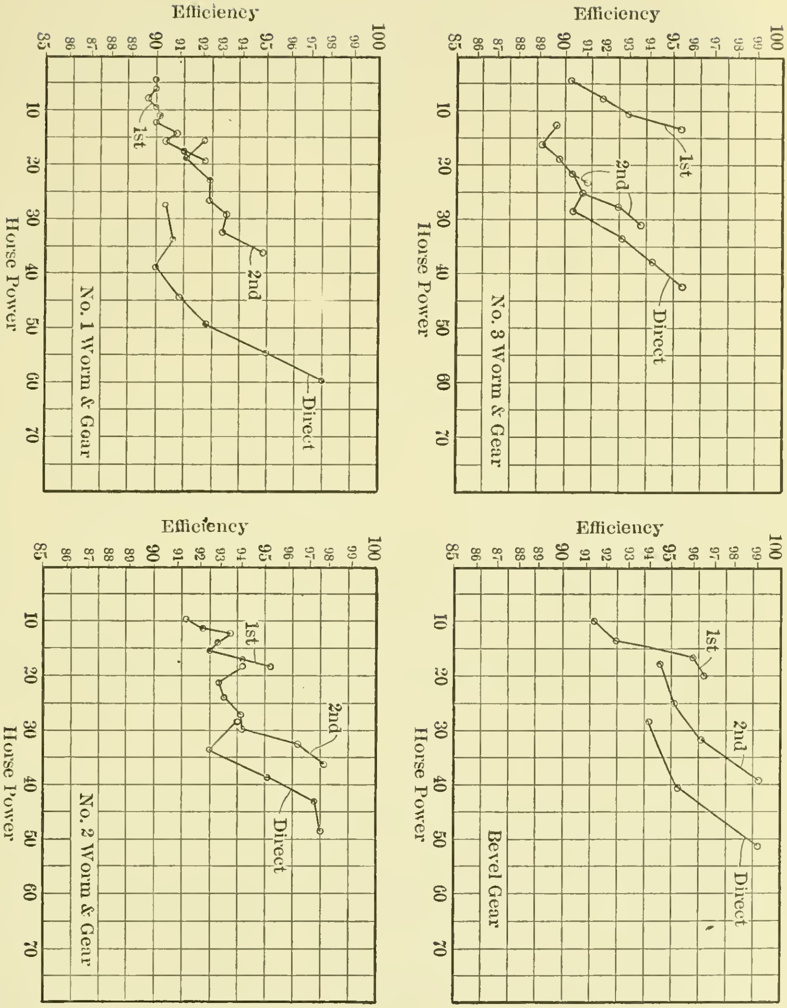


FIG. 13 EFFICIENCY CURVES

constant load was transmitted through the gearing and the temperature of the oil in the gear case and the temperature of the room noted at frequent intervals.

16 From these observations it was found that at the beginning of the run the oil temperature rose rapidly and somewhat irregularly. As the run continued, however, the rise became much more gradual and regular. In the runs where the smaller amounts of power were transmitted a point was reached where the temperature remained constant. This indicated that radiation was sufficient to carry away the heat due to power lost through friction in the gearing, or in other words that the gears would run indefinitely at the load. The heat curves of the No. 1 worm and gear are shown in Fig. 14.

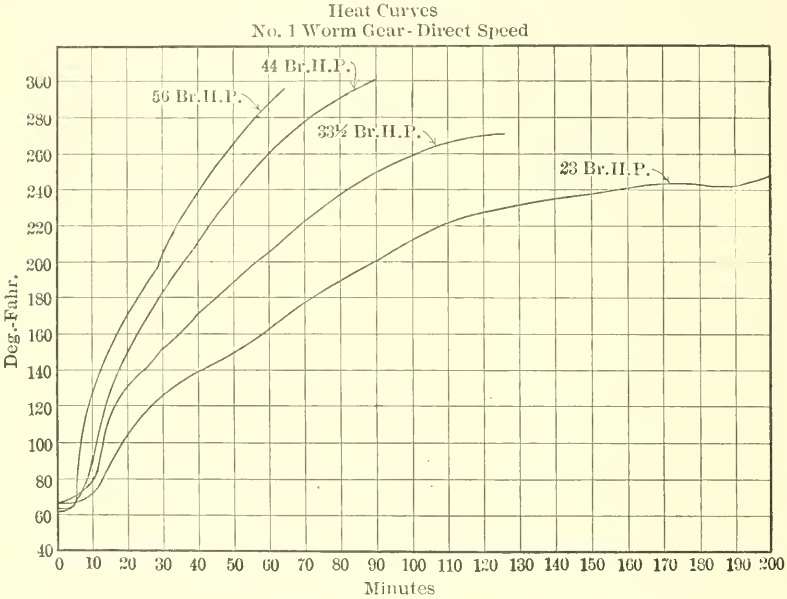


FIG. 14 HEAT CURVES

17 The higher loads indicated were abnormal for the gears under consideration and would not occur in any use to which the gears would normally be put. The fact that these trials continued for from 60 to 80 minutes without failure indicates that the structure is both strong and enduring and that should such temperatures be reached for any accidental reason the gears

would not be destroyed. The result of the trials was of particular interest because of the very high efficiency and carrying capacity of the gears tested.

18 Every possible precaution was taken to secure accuracy in the results, and the high degree of accuracy obtained is due largely to the skill and care of Mr. B. F. Waterman of the Brown & Sharpe Manufacturing Company, under whose personal supervision the apparatus was erected and all the trials were conducted.

MEASUREMENT OF AIR IN FAN WORK

By Charles H. Treat

ABSTRACT OF PAPER

The paper discusses the use of the pitot tube for fan testing with particular emphasis on the correct kind of tube to be used. It describes a simple and accurate method of calibration of impact and static orifices, and the instruments to be used in fan testing, as well as the general method and arrangement of the tests. It shows further the derivation of the formula for friction in test pipe, and the method of correcting test results to standard weight air. The paper further contains proofs that the volume discharged from a fan is the same at all densities, other conditions being the same, and indicates the formulae and calculations used in the test, such as the formula for the determination of air horsepower on the discharge and suction side, efficiency of a fan and its speed. Finally, the question of the average velocity of flow in a pipe is discussed.

MEASUREMENT OF AIR IN FAN WORK

By Charles H. Treat, Honor, Mich.

Junior Member of the Society

Air at fan pressures has come to be used in many fields and for widely different lines of work. For the many occasions where measurement of volume, or of pressure and volume, are required better and more accurate methods are often desired than have been commonly used. The engineer frequently needs something better than the ordinary anemometer to determine the velocity, and a rubber tube connected to a water gage to get the pressure. Especially is this so in fan testing where the volume varies from nothing with closed gate to a maximum with free discharge. The measuring apparatus must be accurate through this wide range.

2 The pitot tube[1] used with suitable pressure measuring devices is particularly well adapted for measuring both pressure and volume, being easily and conveniently arranged for fan testing, also being simple, reliable and accurate.

3 Although the pitot tube has long been known and used there is still much doubt of its accuracy and distrust of results obtained by it. This is especially so for air. The writer believes the trouble in some cases and the resulting general distrust are due to a lack of information as to what forms can *not* be used. In this paper are described what is believed to be an accurate form of tube for this purpose and a sufficiently simple way of using it. The pitot tube will be the only means[2] described, although much of the method employed in the test can be used however the values of pressure and volume were found.

4 This pitot tube work was done from time to time in the Amer-

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[1] For references and a good account of what has been done see The Pitot Tube, W. B. Gregory, Trans. Am. Soc. M. E., vol. 25, p. 184.

[2] Professor Thomas of the University of Wisconsin has developed an excellent device for measuring air or gases known as the Thomas Meter.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

ican Blower Company's fan testing room in Detroit. A little turbine fan was being experimented with by the writer who wanted something smaller and better adapted for use in the small test pipe than the form of tube previously used.[1] The form of pitot tube adopted and which has been in almost daily use for a number of years is shown in Fig. 1. This has been tried out in many ways as by measuring the same air through pipes of different diameter; by measuring in the same pipe with two or more instruments; by measuring the air displaced by a gasometer. These three checks bring in also the important factor in air measuring of getting the mean velocity in the pipe.

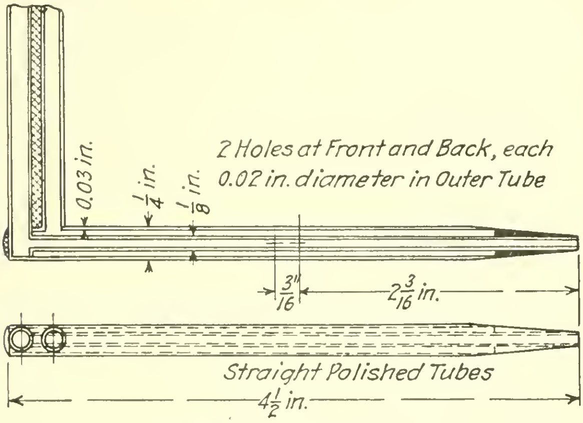


Fig. 1 Standard Pitot Tube

CALIBRATION OF IMPACT AND STATIC ORIFICES

5 In deciding how experimentally to prove the correctness of the orifices of the pitot tube, preference was given to the method which seemed most simple and accurate and which promised to be the best check on work already done.

6 The accuracy of the impact orifice can not be proved by means of a jet of air because the only practical means for determining the velocity at any point in the jet is by the use of the pitot tube. It would scarcely do to test the tube by means of a carrier, such as a locomotive, because of changing air currents. But by attaching the impact tip to the end of a slender hollow arm rotating about a shaft, uniform motion may be obtained and that in air having motion due only to the slender arm through it. If now the shaft be a fan

[1] The pitot tube previously used was the static orifice shown in Fig. 11 at *D*, used with a $\frac{5}{16}$-in. tube pointing into the air current for impact orifice.

shaft and the fan be discharging, air is drawn to the inlet about which the arm is rotating, and by regulating the fan discharge, air can be just drawn away from the path of the tip or, in other words, air flow will be towards the inlet and at right angles to the tip travel instead of with it.

7 We have here a simple means to compare the pressure computed from the relative velocity of tip to air, by the relation $v = \sqrt{2hg}$, with the pressure actually resulting from this velocity. We deal with pressure due to impact only as the atmospheric pressure in the room is on the tip and pressure measuring instrument alike.

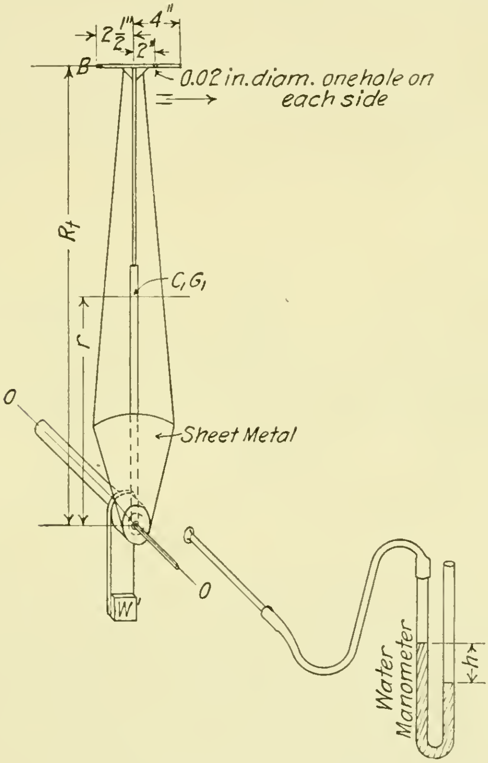


Fig. 2 Diagram for Apparatus for Calibrating Pitot Tube

8 This rotating arm method was the one used for calibrating not only the impact orifice but the static orifice as well. For the static orifice what we wish to know is whether or not it shows the true pressure of the fluid no matter what the velocity may be. As there may be slight radial flow static openings had best be put at the sides (Fig. 2) for then this velocity is only a component of the total veloc-

ity relative to the static opening. If the pressure in the tip be the same as that in the room the pressure at the axis will be less by the centrifugal head in the arm.

9 If the fan be not used the flow caused by the rotating arm is enough to reduce the pressure in the tip from about 2 to 5 per cent, while the component of flow toward the inlet when the fan is used even with the tip pointing to take the resultant maximum velocity will change the pressure less than will show on a slide rule, flow towards the inlet being only enough to take the air away.

10 In older pitot tube work total or impact reading has usually been all right, but static readings have in many cases been inaccurate. A very simple and reliable check on static openings, but which may not be quite accurate, is to try them in a jet of air blowing out freely into the room and observe if the pressure is the same as that in the room, as it should be very nearly even with high velocity. Fig. 3 shows what the static pressures were at the end of a straight 6-in. diameter pipe, the velocity pressure being 2.4 in. head of gasolene, equal to 1.74 in. of water.

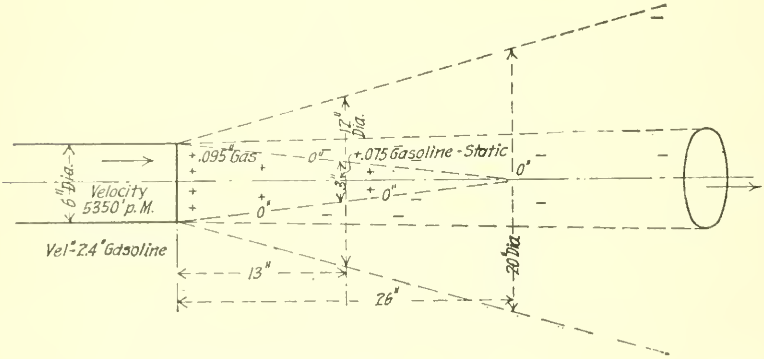


Fig. 3 Static Pressure at Outlet of a Straight Pipe

11 For details of the calibration work, method of computing the centrifugal head, data of tests, etc., see the Appendix. The results are as follows: Working in air instead of water, it is shown that there is no difficulty in getting correct impact or total readings. The form of the tip may be changed from a straight open tube pointing into the current to a small pointed orifice, as in Fig. 1, or may be flat at the tip with a small orifice in the center and still show practically no difference in the readings for the different shapes, showing

that for any ordinary change in the tip no inaccuracy results. The form of impact tip most carefully checked was that shown in Fig. 1.

12 The calibration shows the impact or total reading to be accurate probably to within 1 per cent in the relation $v = \sqrt{2gh}$.

13 A correctly reading static orifice is not so easily obtained. The static of Fig. 1, consisting of clean holes 0.02 in. in diameter in $\frac{1}{4}$-in. tubing $\frac{1}{32}$ in. thick, gave static readings accurate to within less than 1 per cent of pressure due to velocity. A hole $\frac{1}{16}$ in. in diameter in this tubing gave readings considerably off, while a $2\frac{1}{2}$-in. slot $\frac{1}{16}$ in. wide in slightly larger tubing gave static readings as much as 14 per cent too high.

14 An older form of static orifice, shown at *D* in Fig. 4, is fairly accurate, but requires to be set with the plate exactly parallel with the direction of flow. It is often not known just what the direction of flow is, as when there is a slight whirling or spiral motion in the duct. The static openings of the pitot tube shown in Fig. 1 give quite accurate results when 5 deg. out of parallel with flow.

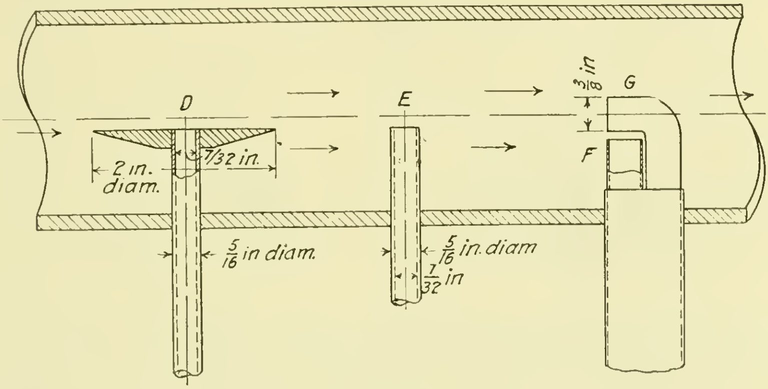


FIG. 4 OLDER FORMS OF STATIC TIPS

15 The open tube *E*, Fig. 4, gives static readings too small by over 50 per cent of the velocity reading. The tube is $\frac{5}{16}$ in. in diameter held at right angles to flow. The static tube *F* in a form of pitot tube that was sold by one of the instrument makers several years ago gives static readings too small by 65 per cent of the true velocity reading. The impact tube gives practically true reading, so that velocity readings by this instrument are much too large.

16 A $\frac{5}{8}$-in. slot 0.01 in. or less wide in the $\frac{1}{4}$-in. tube seemed to give good results with or without inner tube, so also did the large

slots if an inner tube fitted the outer to within 0.01 in. on each side. Wire cloth over the large slots seemed to give good results. These other forms of opening have not been checked and rechecked as have the small 0.02-in. holes. Where the small holes are liable to stoppage, more holes should be used.

METHOD OF TESTING FANS

17 When air is discharged into a pipe by a fan both static and velocity pressures measured from atmospheric are positive, and their sum is shown by the reading in the impact manometer (*B* in Fig. 5). When the air is drawn from a pipe by a fan the static pressure in the pipe is always less than atmospheric; that is, it is negative. The reading in the impact manometer *B* will be also negative, but less than static. Static subtracted from the impact or total gives a positive reading, as it should, for the velocity. As velocity in this pipe is induced by the static pressure, the velocity can never quite equal the static due to the presence of friction.

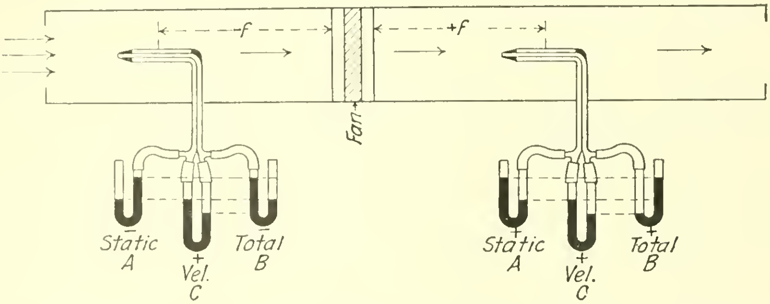


Fig. 5 Arrangement for Measuring Static and Velocity Pressures

18 Figs. 6 and 7 show the method of setting up for a fan test: Fig. 6 for a test on the suction and Fig. 7 for a test on the discharge. Fig. 8 is from a photograph showing the actual arrangement of apparatus under test.

19 By using the pitot tube shown in Fig. 1, where velocity and static are measured practically at the same point, and piping as at *C*, Fig. 5, the pressures other than that due to velocity cancel each other. The possibility of the pressure changing between times of reading static and impact independently is avoided and there is no subtraction to be made. The difference in static pressure between the point of reading in the pipe and atmospheric pressure is read at

A, the total at *B*. The pitot tube should point straight into the current of air to read correctly, either velocity or static.

20 The common pressure measuring apparatus for pressures above 1 in. water gage is satisfactory. For pressures from 1 in. to 2 or 3 ft. of water, manometers are reliable and convenient. Various forms of apparatus are used for measuring pressures of 1 in. or less. As simple and reliable a means as any is probably furnished by an inclined manometer containing gasolene instead of water. The tubes for this manometer should be of the same size, perfectly straight and parallel inside. When so made they can be easily used when inclined as much as 20 in. slant for 1 in. vertical reading, with very accurate results. Water is not a satisfactory fluid in a manometer inclined more than three to one. Gasolene is recommended, as it is not only very active but comes accurately to the same form of surface.

21 Set the back plate by leveling by bubbles in the manometer tubes and then swing the tubes up the angle required to give a good reading, 20 or 10 or 5 to 1. In fan testing for velocity reading at different restrictions two or three slants may be used to cover the wide range in pressure and it is very convenient to cover this wide range with the one gage by the simple device of changing the angle. Good care is necessary to get the angle correct for 20 to 1 and great care for 40 to 1.

22 At 40 to 1 slant for $\frac{1}{2}$ in. velocity reading the velocity is 384 ft., for 0.49 in. reading the velocity is 380 ft., showing that with the slant gage velocities as low as 500 or even 400 ft. per min. may be measured satisfactorily.

23 With skill and care the hook gage gives good results if the pressures do not fluctuate. In air measuring work the pressures usually fluctuate slightly even when care is taken to get the speed of fan as constant as possible. It is desirable to be able to take the readings under these conditions, and this may be done by setting zero-point of the measuring slide at the mean lower level, and at the instant the level comes opposite this point, reading at the upper level.

24 The spacing board for the pitot tube should be placed so that the tube crosses the test pipe centrally. On this board lay off *AB* at the center of travel of the holding block *K* for the tip at extreme movement across pipe. Compute *a*, *b*, *c*, *d* and lay off with *AB* at their center. Inclining manometers are shown for pressure measuring, but other instruments for this purpose may be used if

preferred. After connecting try out the lines of tubing by noting if the liquid lowers in the manometers after pressure has been applied, and make sure that the connections are not clogged. Use care to prevent liquid from being drawn from the gages into the tubing.

25 Make the area of the test pipe the same as the area of the outlet or inlet connections for a fan test. Use well fitted soldered pipe for the test. An ordinary test pipe unsoldered and somewhat carelessly fitted to the fan may leak 5 per cent or more of the air handled at $\frac{1}{2}$ gate, or even at $\frac{5}{8}$ gate, lowering the pressure correspondingly.

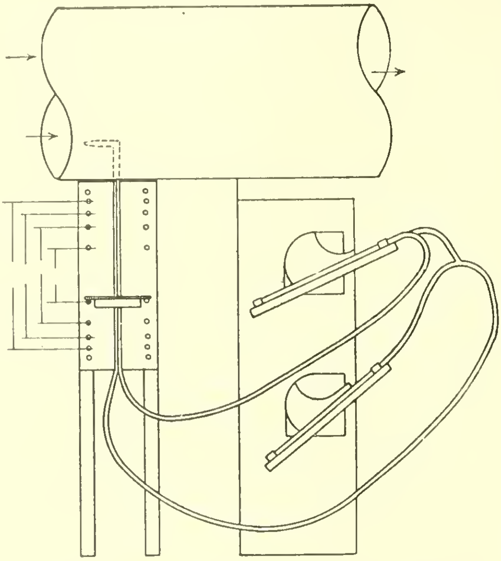


Fig. 6 Correction for Test on Suction

26 It is very essential to determine the relation between volume, the pressures and the power required as the resistance changes. For this purpose it is usual to gate down the outlet for an outlet test or the inlet for an inlet test, using round orifices in thin flat plate. No air should escape except through the orifice. It is usually ample to test with gates 0, $\frac{1}{4}$, $\frac{3}{8}$, $\frac{1}{2}$, $\frac{5}{8}$, $\frac{3}{4}$, and once the area of the orifice. Occasionally a $\frac{7}{8}$ gate is useful. For some tests the $\frac{3}{8}$ and the $\frac{5}{8}$ gates may be omitted. The area of the gate divided by the area of the test pipe is the ratio of opening or the equivalent orifice.

27 The coefficient of efflux is larger for a conical outlet than for an orifice of the same area and is probably greater for a large gate that for a small on the same test pipe. However, a gate, say $\frac{3}{4}$, is very nearly a fixed measure of resistance as above made. Even if it were not there would be no inaccuracy due to this cause as the final curves are ordinarily used, for the *required* ratio of opening is found from the *required* ratio of pressures. The ratio of gate area to full area is often very useful, so that it is wiser to use ratio of opening for abscissa in final curves than some other quantity as volume or ratio of $\frac{V.P.}{D.P.}$

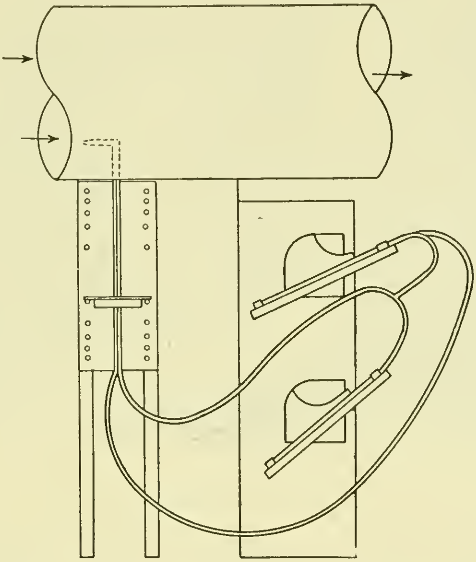


FIG. 7 CORRECTION FOR TEST ON DISCHARGE

28 A convenient log sheet for fan testing is shown in Fig. 9. On this log and on the curve sheets, Figs. 9 and 10, pressures are all expressed in inches of water. The pressure in the impact tube or the total or dynamic pressure is marked Dyn. *W. G.* or simply *D.P.* The pressure in the static orifice (*A* in Fig. 5) is Static *W.G.* or simply *S.P.* The difference between the static and the total (*C* in Fig. 5) is the velocity pressure and is marked Vel *W.G.* or simply *V.P.* The pressure due to a velocity of the air equaling the peripheral velocity of the wheel is the peripheral velocity pressure or *P.V.P.*

29 In making fan tests the speed may not be constant at all times. To correct to a common speed we have relations between speed and pressure and power that have been fully verified by test, the better the tests the better being the check. See curve sheet *A* for a good test showing this point. These relations are reasonable and could probably be established by theory. They are as follows:

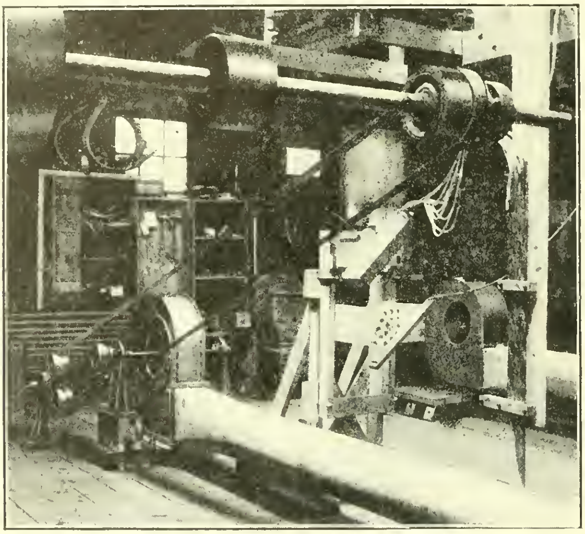


Fig. 8 Fan Testing

30 Under the same conditions of resistance to air flow, density of air and with the same fan

a The volume of air discharged by the fan varies as the r.p.m., that is

$$\frac{\text{vel.}}{\text{Vel.}} = \frac{\text{r.p.m.}}{\text{R.P.M.}}$$

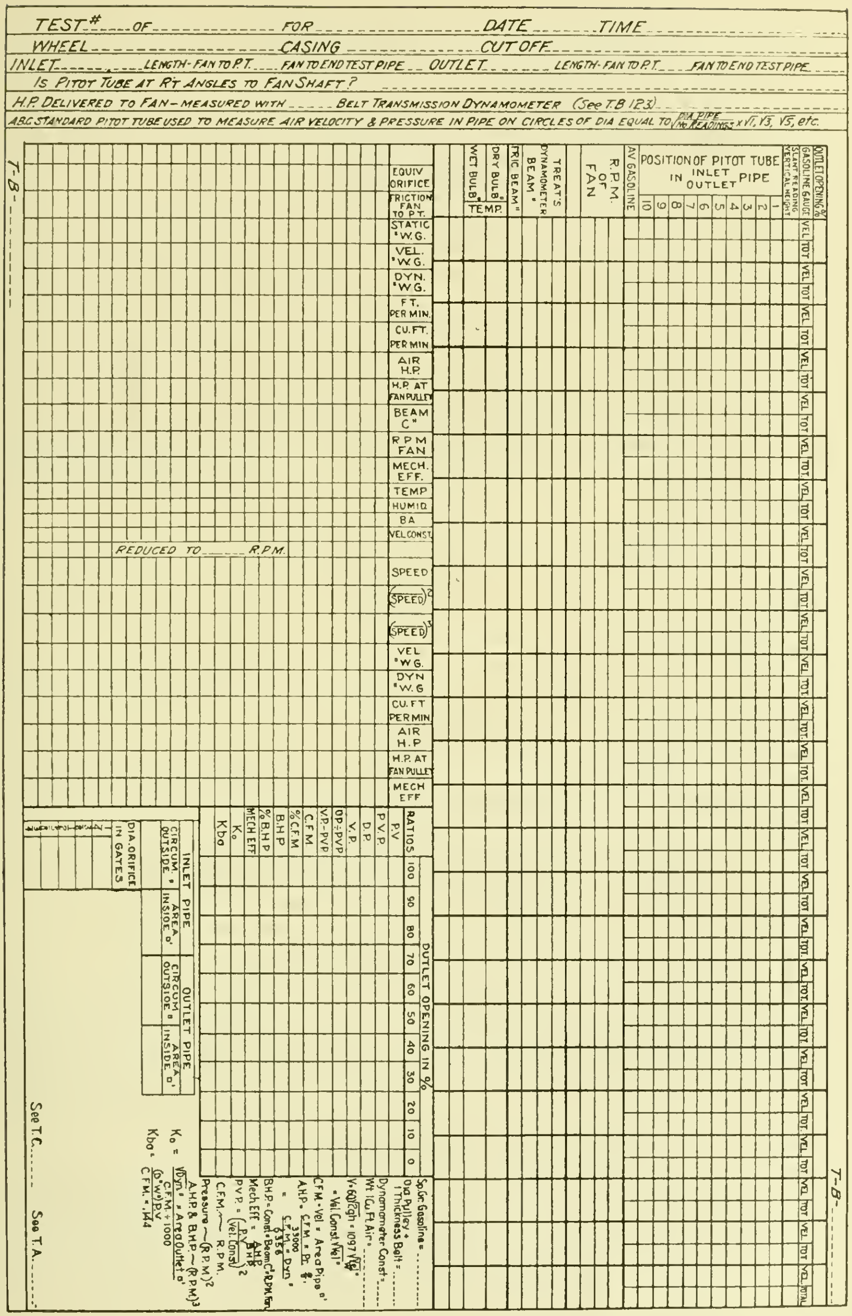
b The pressures produced vary as the square of the r.p.m.

$$\frac{\text{static}}{\text{STATIC}} = \frac{\text{total}}{\text{TOTAL}} = \frac{\text{velocity}^2}{\text{VELOCITY}^2} = \frac{\text{r.p.m.}^2}{\text{R.P.M.}^2}$$

c The horsepower to drive varies as the cube of the r.p.m.

$$\frac{\text{h.p.}}{\text{H.P.}} = \frac{\text{r.p.m.}^3}{\text{R.P.M.}^3}$$

31 *Friction in Test Pipe.* In working up a fan test, a correction for friction in the test pipe must be made to show what the fan alone



TEST # ____ OF ____________ FOR ________________ DATE ______ TIME ______________

WHEEL ________________ CASING ______________ CUT OFF ____________________

INLET ______ LENGTH - FAN TO P.T. ___ FAN TO END TEST PIPE ___ OUTLET ______ LENGTH - FAN TO P.T. ___ FAN TO END TEST PIPE ___

IS PITOT TUBE AT R'T ANGLES TO FAN SHAFT? ____________________

H.P. DELIVERED TO FAN - MEASURED WITH _____ BELT TRANSMISSION DYNAMOMETER (See T.B 123) ____________

ABC STANDARD PITOT TUBE USED TO MEASURE AIR VELOCITY & PRESSURE IN PIPE ON CIRCLES OF DIA EQUAL TO $\frac{\text{DIA PIPE}}{\sqrt{\text{NO. READINGS}}} \times \sqrt{1}, \sqrt{3}, \sqrt{5}$, etc.

T-B- ________

EQUIV ORIFICE

FRICTION FAN TO P.T.

STATIC "W.G.

VEL. "W.G.

DYN. "W.G.

FT. PER MIN.

CU. FT. PER MIN

AIR H.P.

H.P. AT FAN PULLEY

BEAM C"

RPM FAN

MECH. EFF.

TEMP

HUMID.

BA

VEL CONST

REDUCED TO _____ R.P.M.

SPEED

(SPEED)^2

(SPEED)^3

VEL "W.G.

DYN "W.G

CU. FT PER MIN

AIR H.P

H.P. AT FAN PULLEY

MECH EFF

WET BULB° | DRY BULB° — TEMP.

FRIC BEAM"

TREAT'S DYNAMOMETER BEAM"

R.P.M. OF FAN

AV. GASOLINE

POSITION OF PITOT TUBE IN INLET / OUTLET PIPE: 10 9 8 7 6 5 4 3 2 1

SLANT READING VERTICAL HEIGHT

GASOLINE GAUGE

OUTLET OPENING %

VEL TOT VEL TOT … VEL TOTAL

RATIOS: P.V, P.V.P., D.P., V.P., OP ÷ PVP, V.P.-PVP, C.F.M, % C.F.M, B.H.P, % B.H.P, MECH EFF, K_o, Kbo

OUTLET OPENING IN %: 100 90 80 70 60 50 40 30 20 10 0

DIA. ORIFICE IN GATES: 1 2 3 4 5 6

INLET PIPE: CIRCUM. OUTSIDE = | AREA INSIDE □"

OUTLET PIPE: CIRCUM. OUTSIDE = | AREA INSIDE □"

Sp. Gr. Gasoline = ...

Dia. Pulley + 1 Thickness Belt = ...

Dynamometer Const = ...

Wt 1 Cu. Ft. Air = ...

$V = 60\sqrt{2gh} = 1097\sqrt{\frac{\text{Vel}}{\text{Wt}}}$ = Vel. Const $\sqrt{\text{Vel}}$ =

C.F.M. = Vel × Area Pipe □'

$\text{A.H.P.} = \frac{\text{C.F.M.} \times \text{Pr.}}{3300} \times \frac{5}{8}$.

$= \frac{\text{C.F.M.} \times \text{Dyn}}{6356}$ =

B.H.P. = Const × Beam C" × R.P.M. Fan

$\text{Mech Eff} = \frac{\text{A.H.P.}}{\text{B.H.P.}}$

$\text{P.V.P.} = \left(\frac{\text{P.V.}}{\text{Vel. Const}}\right)^2$

C.F.M. ∼ R.P.M.

Pressure ∼ (R.P.M.)^2

A.H.P. & B.H.P. ∼ (R.P.M.)^3

$K_o = \frac{\sqrt{\text{Dyn}}"}{\text{C.F.M.} \div 1000} \times \text{Area Outlet}$ □'

$\text{Kbo} = \frac{(\text{□}" \text{w}) \text{P.V}}{\text{C.F.M.} \times .144}$

See T.C. .... See T.A. ....

T-B- ....

FIG. 9 LOG SHEET FOR FAN TEST

is really doing. The friction pressure or pressure to overcome this friction may be 10 to 15 per cent of the total at wide open gate and 3 to 4 per cent at $\frac{1}{2}$ gate.

32 For ordinary round galvanized pipe a simple and quick way to find this friction is from the formula

$$F=\frac{\text{Vel.}''\times L'}{40\times D'} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [1]$$

where F = friction in inches of water L and D = length and diameter of round pipe. Or in a form more easily remembered, the length of pipe in which friction inches equal velocity inches pressure is 40 diameters long. This rule is applicable for air under ordinary conditions of temperature, pressure and humidity, and values by it check closely with those given by Kent.

33 For smooth straight pipe instead of 40 times the diameter it may be 60 or more and for rough beaded pipe the figure will be less that 40.

34 A general equation[1] for friction is

$$P=\frac{fsyv}{A2g}$$

for any pipe, and

$$P=\frac{4\,fyLv}{2gD} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [2]$$

for round pipe

y = weight of 1 cu. ft. of air
P = pressure lb. per sq. ft.
L and D = length and diameter of pipe in ft.
f = coefficient of friction
v = velocity in ft. per sec.
s = perimeter of section of pipe

41 If we substitute in [2] the value 0.0067, that being the coefficient of friction in ordinary piping, for coefficient of friction and 0.0715 as the weight of air we can derive [1]. It is interesting to note that 0.0067 is given as the coefficient of skin friction for water of rather rough surface. Is the coefficient for air and water the same? Why should they not be the same?

42 *Standard Weight Air.* It is important to have a standard weight of air at which fan performances may be compared. The density, while ordinarily a small factor, affects the results consider-

---

[1] See Church's Hydraulics.

ably, if extremes are great, as with heated air or with cold out-door air. It affects the velocity pressure, the total pressure and the horsepower.

43 As correction to a standard involves some labor it is desirable to choose for a standard a mean weight so that the original data will be close enough without correction in many cases, or so that the conditions may be kept such as will give standard weight. Another reason for the mean or ordinary weight is that such data as taken may be compared with previous tests, also that ordinary rough uncorrected tests may be compared with standard tests.

44 As fan tests are apt to be conducted indoors with warmed air the weight on the average for the year in the United States will not be far from 0.0715 lb. per cu. ft. This in fact is the average weight for a large number of tests extending over a number of years. This weight gives a velocity constant of 4100 or

$$\text{vel.} = 4100 \sqrt{\text{vel.}''}$$

There is some advantage in a short figure in computations from the final curves.

45 To correct to standard air, multiply the velocity pressure, total pressure and the horsepower of the actual test by $\dfrac{0.0715}{\text{wt. 1 cu. ft.}}$ The velocity will be the same as though found at the actual velocity pressure, but the static and total pressure change as the density, changing the air horsepower and the actual horsepower in proportion.

45 A fan discharging under constant conditions as to speed and resistance, will maintain a constant head and a constant velocity of discharge with changing density. That this is so is apparent from the following:

47 In a pipe system supplied by a fan running at a constant speed all the resistance may be classed as friction of various kinds. The impeller sets up heads, due to centrifugal force and to motion imparted to the air, which total head is resisted by internal and pipe frictions.

$$\text{The net head } h_0 + h_v + h_f + h_{f1} = \frac{V^2}{2g} \dots\dots\dots\dots [3]$$

48 Density of the fluid does not affect these heads, and the net head for one density will be the head for all. The velocity will be the same for all from the law of falling bodies.

Centrifugal head

$$P = h_c y = \frac{wv^2}{gr} = \frac{2ryv^2}{gr} \quad\text{.................... [4]}$$

$$h_c = \frac{2rv^2}{gr} \quad\text{........................ [5]}$$

Head due to motion

$$h_v = \frac{v^2}{2g} \quad\text{.............................. [6]}$$

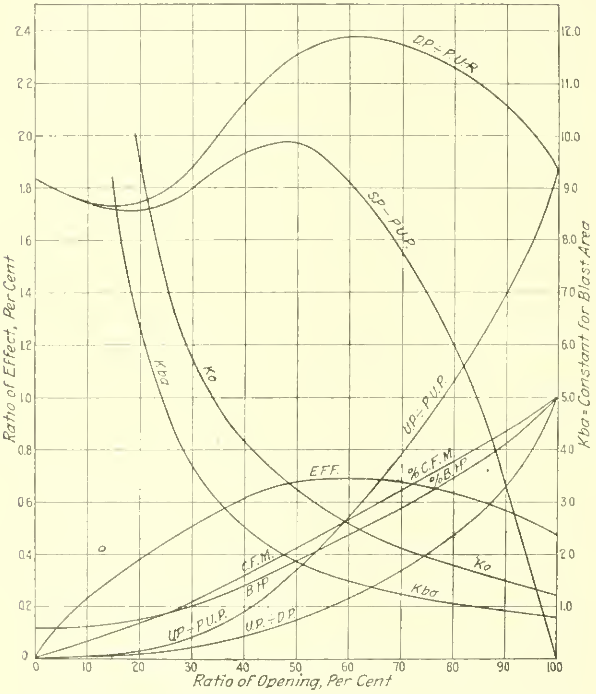


Fig. 10 Characteristic Curves of a Fan Test

Heads due to friction pressure

$$P = h_f = \frac{fSyv^2}{A2g} \quad\text{........................ [7]}$$

and

$$h_f \text{ or } h_{f1} = \frac{fSv^2}{A2g} \quad\text{........................ [8]}$$

h_c = head due to centrifugal force.

h_v = head due to velocity and velocity change

h_f = head due to friction in case
h_{f1} = head due to friction in pipe
P = pressure per sq. ft.
y = density or weight of 1 cu. ft. air

49 From this it follows that the volume discharged from a fan is the same at all densities, other conditions being the same. This is probably true for all fluids, since the coefficient of friction seems to be the same.

50 A thoroughly good way of making the results of a fan test available and useful is to plot characteristic curves, as in Fig. 10.

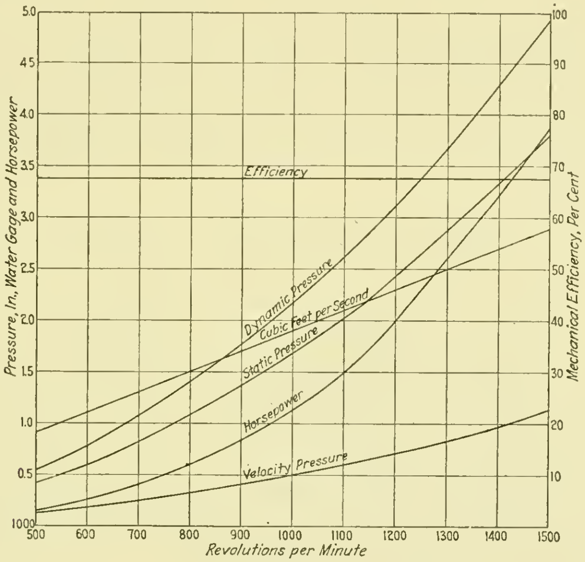


FIG. 11 PRELIMINARY CURVES OF FAN TEST

51 The original readings are first corrected to one common speed, if the speed has varied through the test, as it is likely to do from belt slip. These data at a common speed are then plotted on a preliminary curve sheet similar to the curve sheet in Fig. 11, but with the ratio of opening plotted as abscissa instead of r.p.m. On this sheet correction can be made to standard weight air and for pipe friction. This preliminary set of curves will usually show up a mistake in observation or in computation. From this curve sheet practically any fan problem may be solved for fans of that kind.

Of course the other fans of the line must be kept in proportion to the one tested to get the result.

FORMULAE AND CALCULATIONS USED IN TESTS

52 Bernouilli's theorem states that the sum of all the pressures of a fluid at one section of a system of smooth piping equals the sum of all the pressures at any other section, pipe friction being cared for by the difference in friction pressures.

53 If a fluid motor be introduced between the two sections where pressures are taken it will create a difference in pressure. This difference in pressure times the volume of fluid passing through the motor times the constants for heaviness of fluid, area, etc., gives the horsepower produced by the motor, or given to it if power is being taken from the fluid. But for the pressure caused by the motor, the sums of pressures at the two sections would be equal.

$$\frac{\text{cu. ft.}}{\text{const.}}\left[-\left(v''+(-s''-f'')\right)+(V''+S''+F'')\right]=\text{h.p.}$$

$$\frac{\text{cu. ft.}}{\text{const.}}\left[V''+S''+F''+s''+f''-v''\right]=\text{h.p.}\ldots\ldots\ldots\ldots[9]$$

$$\frac{\text{cu. ft.}\times\text{difference in pressure}\times 62.3}{33000 \qquad\qquad 12}=\text{air h.p.}$$

$$\frac{\text{cu. ft.}\times\text{difference in pressure}}{6356}=\text{air h.p.}$$

v'' = velocity, in. suction

s'' = static, in. of water

f'' = friction, in. from fan inlet to measurement on suction

V'' = velocity, in. water in discharge

S'' = static, in. water in discharge

F'' = friction, in. at discharge from point of measurement to fan (see Fig. 5)

When the friction heads F'' and f'' are considered as belonging to the air motor, then

$$\frac{\text{cu. ft.}}{\text{const.}}\left[V''+S''+s''-v''\right]=\text{air h.p.}\ldots\ldots\ldots\ldots[10]$$

54 When the discharge pipe and suction pipe are of the same diameter $V''-v''=0$ and equation [10] becomes

$$\frac{\text{cu. ft.}}{\text{const.}}\left[S''+s''\right]=\text{air h.p.}\ldots\ldots\ldots\ldots\ldots\ldots[11]$$

55 It is quite usual in buying and selling fans to understand that a certain effect on discharge shall be produced; that is, such a volume and pressure in discharge pipe.

56 When the work sold for is to be done on the discharge side the inevitable friction loss in drawing air through the inlets is not to be measured in. It is not wanted although as in the case of some single inlet fans it may be a very appreciable factor. Large inlets reduce it. For this case,

$$\frac{\text{cu. ft.}}{\text{const.}}\left[V''+S''+F''\right]=\text{air h.p. on discharge side}\ldots\text{[12]}$$

and

$$\frac{\text{air h.p.}}{\text{h.p. to fan}}=\text{Efficiency for discharge side}$$

For a fan that does its work on the suction side with a short outlet on the discharge the same size as the inlet, a usual arrangement, we have

$$\frac{\text{cu. ft.}}{\text{const.}}\left[V+0+0+s''+f''-v''\right]=\text{air h.p. from [9], or when } V''=v''\ldots\text{[13]}$$

$$\frac{\text{cu. ft.}}{\text{const.}}\left[s''+f''\right]=\text{air h.p. on suction side}\ldots\ldots\ldots\text{[14]}$$

If it is agreed to use an expanding or enlarged outlet such as the chimney of mine fans, equation [13] with V'' due to velocity at the outlet of the chimney should be used.

57 It might be agreed to sell for work on the suction side alone, then $\frac{\text{cu. ft.}}{\text{const.}}\left[s''+f''-v''\right]=$ air h.p. and it would be up to the fan maker to reduce V'' as much as possible.

58 When a fan is to do work on both sides the equations [9], [10] and [11] ought to be used to measure the air h.p.

59 A cone on outlet will not increase the efficiency of a fan but will decrease work to be done, that is, the same volume of air can be moved through the same resistance by less power, the speed being a little slower.

60 For air measuring work the formula $v=\sqrt{2gh}$ may be written

$$V=1097\sqrt{\frac{\text{vel.'' W. G.}}{\text{wt. cu. ft. air}}}\ldots\ldots\ldots\ldots\ldots\text{[15]}$$

When head due to velocity is expressed in its equivalent inches water gage readings, and V = Vel. in ft. per min.

$V = 4100\sqrt{V''\ \text{W.G.}}$ for standard weight air of 0.0715 lb. per cu. ft.

AVERAGE VELOCITY OF FLOW IN A PIPE

61 The velocity is least at the surface of the pipe because of skin friction. In a square pipe the mean of the velocities and of the pressures taken at the center of a large number of equal squares into which the pipe is divided, is the average velocity and pressure in the pipe. In the case of a round pipe, divide into equal annular areas, and for each read at a point where the area inside equals the area outside (see Fig. 12). A convenient rule for obtaining diameters of circles is as follows:

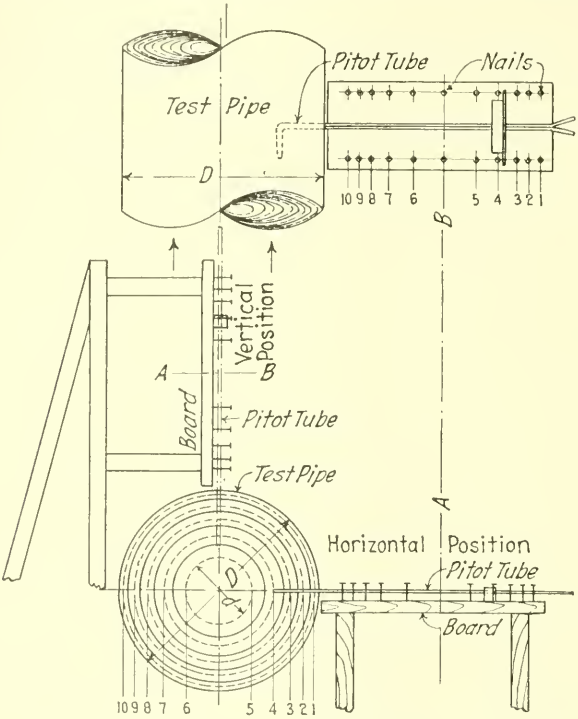


Fig. 12 Setting Pitot Tube

62 Divide the diameter of the air pipe by the square root of twice the number of readings to be made from the circumference

to the center for the diameter of the smallest circle. Multiply this reading by $\sqrt{3}$, $\sqrt{5}$, etc., for diameters of larger circles. For example, for $48\frac{7}{16}$ diameter of pipe and eight readings $\frac{48\frac{7}{16}}{\sqrt{2\times4}}=17\frac{1}{8}\times\sqrt{3},\ \sqrt{5},$ etc. $=29\frac{5}{8}$, $38\frac{1}{4}$, etc. This is done very readily on the slide rule by dividing the diameter pipe found on the lower scale by the square

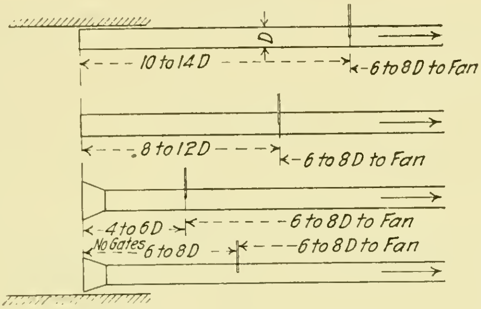


Fig. 13 Length for Test Pipe on Suction

TEST IN DISCHARGE PIPE 20 DIAMETERS LONG. READINGS ⅝ IN. LENGTH FROM FAN

Position of pitot tube across pipe..	1	2	3	4	5	6	7	8	9	10
Velocity pressure in test pipe, in. .	0.730	0.885	0.990	1.025	1.060	1.075	1.055	1.022	0.925	0.680
Velocity, ft. per min.	3503	3850	4080	4150	4222	4260	4220	4150	3950	3380

Mean velocity pressure in test pipe, 0.9447 in.; mean velocity, 3976.5 ft. per min.

$4100\times\sqrt{0.9447}$ in. $=3985$ ft.

root of the numbers found on the slide. Without moving the slide multiply by $\sqrt{3}$, $\sqrt{5}$, etc., which roots are also found on the slide.

63 This rule may be demonstrated as follows: Divide the annular rings in two by a vertical line through the center. (See Fig. 12.)

$$\text{Total area}=A=\frac{\pi}{4}D^2$$

Circle d contains two of the equal areas of the 20 into which D is divided.

$$a = \text{area of small circle} = \frac{\pi d^2}{4}$$

$$\text{Unit area } \frac{a}{2} = \frac{A}{20}$$

$$A = \frac{a}{2} \times 20 = 10a,$$

$$10d^2 = D^2$$

since the area of a circle varies as the square of the diameter.

$$d^2 = \frac{D^2}{10}$$

$$d = \frac{D}{\sqrt{10}}$$

64 Since in equation [15] the velocity changes as the square root of the velocity water gage the velocities should be computed for each unit area and those averaged. The error is small, however, if the velocity be computed from the average of the velocity inches, ordinarily being less than 1 per cent from this cause.

65 There are usually irregularities of flow in air entering a pipe and if it is the discharge from a fan the irregularities may be considerable; if an inlet pipe there will be less and if fitted with cone inlet there will be considerably less irregularity of flow. This irregularity becomes less away from the pipe inlet and in 8 to 12 diameters in the case of fan discharge and in straight pipe good pitot tube readings may be taken. (See Fig. 13.)

66 For accurate measurement it is recommended that both vertical and horizontal readings across the pipe be taken rather than either alone, especially if there has been a bend in the pipe or if the pipe is the discharge from a fan.

APPENDIX

CALIBRATION OF STATIC AND IMPACT ORIFICES

67 This calibration was made by revolving the various tips at the end of a slender arm and noting the speed and the pressures at the axis. The arm was made of brass tubing $\frac{3}{16}$ in. and $\frac{3}{8}$ in. in diameter, sweated together. Stay wires were used to give it rigidity, as shown in Fig. 2, and the tubing was fastened to its countershaft by the set screw of the collar through which the larger tube fastens. The different tips tried were soldered to the end of this arm. A very small air-tight connection or stuffing box at the axis was afforded by a $\frac{1}{8}$-in. brass tube projecting from the larger tube along the axis and having a ring of greased candle wicking on it against which the cone end of a larger tube was pressed, the small tube revolving in it. The arm was counterweighted at *W*. Care was taken to press on the outer tube only hard enough to keep the connection tight.

68 The revolutions were counted by a hand-speed counter at the other end of the shaft, usually for two or three minutes. The length of arm was measured from axis to center of orifice by steel scales. The shaft was driven by an automatic engine having no other load so that the speed was very steady and uniform.

69 Pressure was taken by a vertical water manometer of large tube area (0.45 in. inside diameter), in most of the runs. An inclined gasolene manometer was used, in those of small head the reading being corrected to the head of water. Temperature and humidity were taken in the room not far from the arm.

70 The $\frac{1}{4}$-in. tubes at the end of the arm which were used in all the runs except the last four projected forward from the arm about $3\frac{1}{4}$ in. and backward about $2\frac{1}{2}$ in. The static holes in the sides were about 2 in. ahead of the arm. Different tubes were used but were adjusted alike, being soldered to the corner braces and pipe.

71 Air being elastic, that portion nearer to the shaft presses, due to centrifugal force, on the air further out in the tube compressing it slightly. Therefore the outer portion of the air in the tube is slightly more dense than the inner, and this varying density makes it necessary to go to calculus for a solution.

72 For an inelastic fluid the computation would be very much simpler, and for a rotating column of unit area it would be as follows:

$$\text{Centrifugal force (pressure per sq. ft.)} = \frac{wv^2}{gr} = \frac{w(\pi 2rn)^2}{gr} = \frac{2ry(\pi 2rn)^2}{gr}$$

w = total weight of fluid in lb.
v = velocity in ft. per sec.
g = gravity
r = radius to center of gravity
y = weight per cu. ft. of fluid
n = revolutions per second

DIFFERENCE IN PRESSURE BETWEEN THE ENDS OF A REVOLVING RADIUS OF ELASTIC FLUID

73 Assume
PV = constant
p = pressure in lb. per sq. ft. at any intermediate point, variable
P_o = pressure in lb. per sq. ft. at axis due to the atmospheric pressure
P_t = pressure in lb. per sq. ft. at the free end of the radius—at tip
W = weight per cu. ft. of elastic fluid, variable
W_o = weight per cu. ft. of elastic fluid at atmospheric pressure
N = r.p.m.
R = radius in ft.
R_o minimum radius
R_t = maximum radius
$B\alpha$ = barometer at time of experiment

74 For a very short radius of elastic fluid

$$p = \frac{WR\left(2\pi\frac{R}{2}\frac{N}{60}\right)^2}{g \times \frac{R}{2}} = \frac{2\pi^2WR^2N^2}{g \times 3600}$$

$$dp = \frac{2 \times 2\pi^2WN^2}{g \times 3600} RdR$$

and as

$$\frac{W}{p} = \frac{W_o}{62.3 \times \frac{B\alpha \times 13.61}{12}}$$

$$dp = \frac{4\pi^2 \times 12W_oN^2pRdR}{g \times 3600 \times 62.3 \times 13.61 \times B\alpha} = C \times pRdR$$

$$\frac{dp}{p} = CRdR$$

$$\int_{P_o}^{P_t} \frac{dp}{p} = C\int_{R_o}^{R_t} RdR$$

integrating

$$\log_e \frac{P_t}{P_o} = \frac{C}{2}(R_t^2 - R_o^2) + K$$

when $R_t = R_o$, $P_t = P_o$ and $\frac{P_t}{P_o} = 1$. Log of $1 = 0$ and for this $(R_t^2 - R_o^2) = 0$, K must $= 0$. As $\log_e =$ common log $\times$ 2.302585 we have

$$\text{common log } \frac{P_t}{P_o} = \frac{C}{2 \times 2.302585}(R_t^2 - R_o^2) \ldots\ldots\ldots\ldots \quad [18]$$

$\frac{P_t}{P_o}$ is the ratio of pressure at the ends of radius and may be expressed in any units of pressure or head.

$$\frac{C}{2 \times 2.302585} = \frac{2 \times \pi^2 \times 12 \times W_o \times N^2}{2.302585 \times g \times 3600 \times 62.3 \times 13.61 \times B\alpha} = 0.0000010479 \frac{W_o \times N^2}{B\alpha}$$

No.	log	No.	log
2	0.301030	2.302585	0.362210
π^2	0.4971498	32.16	1.507316
	0.4971598	3600.	3.556303
12	1.079181	62.3	1.794488
	2.3745106	13.61	1.133858
	8.354175		8.354175
0.0000010479	......$\bar{6}$.0203356		

Equation [18] with numerical constants is

$$\text{common log } \frac{P_t}{P_o} = \log \frac{\text{in. mercury}}{B\alpha} = 0.0000010479 \frac{W_o N^2 R_t^2}{B\alpha}$$

75 *Impact Orifice.* When the pressure at the axis is atmospheric, or zero on the manometer then the common log of pressure at the tip in inches of mercury is

$$0.0000010479 \frac{W_o N^2 R_t^2}{B\alpha} + \log B\alpha \ldots\ldots\ldots\ldots\ldots\ldots \quad [19]$$

The number corresponding to log $B\alpha$+the numerical factor, the second part of equation [19] less the barometer pressure is the difference in pressure between the two ends, or the centrifugal head.

76 *Static Orifice.* If a static orifice is correct the pressure inside the tip = pressure outside; or atmospheric, at all velocities of air past; then suction k on the manometer at the axis = centrifugal head in the arm. For this case

$$\text{log pressure at tip} = 0.0000010479 \frac{W_o N^2 R_t^2}{B\alpha} + \log (B\alpha - K) \ldots\ldots \quad [20]$$

and the number corresponding to the second part of equation [20] less the pressure at the axis $(B\alpha - K)$ is the centrifugal head, K and $B\alpha$ of course being expressed in the same unit of pressure as mercury.

77 *Impact.* Of the runs for impact the first nine were with a fan drawing away the disturbed air. Of these the ones with full gate were affected only very slightly, if at all, by the air motion due to the rotating arm. This point was determined by testing with smoke. The air travel toward the inlet was probably never anywhere more than 300 ft. per min. At the smaller gates there was, however, some motion of the air with the arm so that the runs with full gate deserve more consideration that those at small gate.

TABLE 1 PITOT TUBE CALIBRATION

Test No.	R.P.M.	A Manometer Reading, In. Water	B Centrifugal Head, Computed	$A+B$ Pressure in Tip Back of Orifice	C Pressure Due to Velocity of Tip in Still Air, In. Water	Pressure in Tip ÷ Pressure Due to Velocity $A+B \div C$	Weight of 1 Cu. Ft. of Air	Humidity Temperatures		Barometer	Length of Arm Axis to Tip, In.	Date (1908) and Time	Remarks
								Dry	Wet				
1	129¾	00	0.489	0.489	0.4888	1.000	0.07247	75°	60°	29.34	41.656	10/15	Impact orifice; ½ gate on fan
2	328½	0	3.112	3.112	3.095	1.003	0.07247	75°	60°	29.34	41.656	a.m.	Impact orifice; full gate
3	388	0	4.4	4.4	4.371	1.004	0.07247	75°	60°	29.34	41.656	a.m.	Impact orifice
4	0 to 480	0	.....	......	......	....	0.07247	75°	60°	29.34	41.656	a.m.	Impact orifice; slow up to full speed. Air motion slow and nearly radial to shaft
5	361	a little +	3.727	3.727	3.709	1.005	0.07192	79°	64°	29.36	41.656	2 p.m.	Impact orifice; air currents change slightly about inlet
6	482	a little −	6.665	6.665	6.612	1.008	0.07192	79°	64°	29.36	41.656	....	Impact orifice
7	430¾	−⅛ to −7/32	5.31	5.18	5.27	0.984	0.07182	79°	64°	29.32	41.656	10/14	Impact orifice, but ½ gate
8	414	0	4.905	4.905	4.870	1.006	0.07182	75°	64°	29.32	41.656	10/14	Impact orifice, but full gate
9	414	−⅛	4.905	4.78	4.870	0.983	0.07182	75°	64°	29.32	41.656	10/14	Impact orifice, but ¼ gate
10	464	−3/16	.....	......	......	....	0.06997	92°	78°	29.40	41.62	7/6	3/32 in. impact orifice; Std.
11	464	−7/32	.....	......	......	....	0.06997	92°	78°	29.40	41.62	p.m.	5/16 in. diameter; flat point
12	506.5	−0.23	.....	......	......	....	0.07152	82°	69½°	29.43	41.7	8/15	¼ in. tube open
13	505.5	−0.24	.....	......	......	....	0.07177	78½°	67½°	29.33	41.7	8/18	Std. impact orifice
14	504	−8¼	7.185	−1.065	÷ B	−0.148	0.07202	78°	67°	29.41	41.80	8/26	Back end of tip open; ¼ in. tubing tip square
15	506	−9⅜	7.21	−2.165	÷ B	−0.305	0.07202	78°	67°	29.41	41.80	3.50 p.m.	Same; but with cord around tube at tip 11/32 in.

16	508	−8.515	7.28	−1.235	÷ *B*	−0.1695	0.07202	78°	67°	29.41	41.80	....	3/16 in. tube square as was 1/4 in.
17	503.5	−8.44	7.19	−1.25	÷ *B*	−0.174	0.07202	78°	67°	29.41	41.80	....	1/4 in., tube, but beveled 45 deg.
18	507	−0.328	7.4	7.072	7.02	−1.008	0.07202	78°	67°	29.41	41.67	4.00 p.m.	Std. impact; air travel with arm; 245 ft. by anemometer
19	142	−0.562	0.572	0.010	÷ *B*	+0.0175	0.07199	77½°	65½°	29.34	41.625	7/6	Std. static—holes 0.02 in. in diameter
20	193.5	−1.047	1.062	0.015	1.065	+0.014	0.07199	77½°	65½°	29.34	41.625	p.m.	Std. static—holes 0.02 in. in diameter
21	242.2	−1.6875	1.667	−0.020	÷ *B*	−0.012	0.07199	77½°	65½°	29.34	41.625	p.m.	Std. static—holes 0.02 in. in diameter
22	508	−7.32	7.27	−0.05	÷ *B*	−0.0069	0.07208	76°	64¼°	29.29	41.62	8/20	No leak at axis. Std. static—holes 0.02 in. in diameter No inner tube.
23	511	−7.34	7.375	+0.035	÷ *B*	+0.0048	0.07208	76°	64¼°	29.29	41.62	5.15 p.m.	Std. static—holes 0.02 in. in diameter
24	512	−7.36	7.4	+0.04	÷ *B*	+0.0054	0.07208	76°	64¼°	29.29	41.62	....	Std. static—holes 0.02 in. in diameter
25	513	−7.405	7.385	−0.02	÷ *B*	−0.0028	0.07208	76°	64¼°	29.29	41.62	....	Std. static—holes 0.02 in. in diameter
26	515	−7.46	7.48	+0.02	÷ *B*	+0.0028	0.07208	76°	64¼°	29.29	41.62	....	Std. static—holes 0.02 in. in diameter
27	505.5	−7.125	7.23	+0.105	÷ *B*	+0.014	0.07216	77°	67°	29.41	41.67	8/26	Std. static—2 holes 0.026 in. in diameter and 2 −0.02 in. and
28	507.5	−7.156	7.28+	+0.124	÷ *B*	+0.017	0.07216	77°	67°	29.41	41.67	2 p.m.	Std. static inner tube—⅛ in. in diameter
29	507.3	−7 3/16	7.28	+0.092	+ *B*	+0.0126	0.07216	77°	67°	29.41	41.67	....	Std. static inner tube used in Nos. 27 to 33
30	504.6	−7.125	7.2	+0.075	÷ *B*	+0.010	0.07202	78°	67½°	29.41	41.67	....	2 holes 0.02 in. in diameter and inner tube ⅛ in. in diameter
31	506.5	−7.156	7.255	+0.099	÷ *B*	+0.0136	0.07202	78°	67½°	29.41	41.67	....	Same as last; very small leak stopped
32	504	−7.156	7.19	+0.034	÷ *B*	+0.0047	0.07202	78°	67½°	29.41	41.67	....	
33	506	−7.187	7.25	+0.063	÷ *B*	+0.0087	0.07202	78°	67½°	29.41	41.67	3 p.m.	
34	463	−5.396	6.035	+0.639	÷ *B*	+0.106	0.07050	86°	70°	29.20	41.76	7/10	⅛ in. pipe about 0.28 in. diameter inside with 2 slots opposite 1/16 in. x 2½in.—static orifice—mean of 4 tests
35	511.2	−7.23	7.33	+1.05	÷ *B*	+0.0143	0.07090	81°	75°	29.21	41.76	8/13, a.m.	Above, with ¼ in. tube inside central
36	501.5	−6.65	7.08	+0.43	÷ *B*	+0.061	0.07057	84½°	75°	29.24	41.76	8/13, 3 p.m.	Above, with inner tube 0.02 in. smaller
37	502.5	−6.4	7.10	+0.70	÷ *B*	+0.099	0.07057	84½°	75°	29.24	41.76	8/13, 3.30 p.m.	Above. No inner tube—same as No. 34

78 Another minor source of trouble was the tendency of the stuffing box to heat the air in the tube slightly. This was largely overcome by the very small size and by care to use soft grease freely on the wicking, also by not pressing the outer tube against the wicking harder than necessary. Between runs outside air was allowed to replace the warmed air, but doubtless the results are affected very slightly by the air in the tube being a little warmer than that in the room. The brass tube of the arm always felt as cool after a run as before. Warming of the brass tubes of the stuffing box to as slight an extent as 5 is noticeable to touch at these temperatures.

79 An increase of 10 deg. in temperature in run No. 8 without change of moisture in the air in arm, changes the computed centrifugal head from 4.905 in. at room temperature to 4.825 in.

80 Of the first nine runs only No. 6 at the highest speed shows a negative pressure in the water manometer used, when the fan was fully open. With the smaller gates the readings were slightly negative. The mean of these runs is 0.9991 for the pressure in the tip divided by the pressure due to velocity. However, considerably more weight must be given the runs with full gate.

If the runs with the smaller gates be left out then the pressure in the tip exceeds that due to velocity by an amount that would have been noticed and measured, for the water levels in the manometer used were steady and the conditions good. It is believed that the slight rise in temperature of the air in the tubes accounts for the observed pressure being slightly less than the computed.

81 Runs Nos. 10 to 18 inclusive were all made without the fan to draw the disturbed air away. Nos. 10 and 11 show a comparison of a flat impact point $\frac{5}{16}$ in. in diameter with a hole in the center about $\frac{1}{16}$ in. in diameter with the same tip used in the first runs, marked Std. Nos. 12 and 13 compare a $\frac{1}{4}$-in. open brass tube with end filled square with Std.

82 Nos. 14 to 17 were with the back end of the tip open (see *B* in Fig. 2), using four forms of opening. No. 18 is given to show about what air motion the arm set up for these runs. These readings are given because nearly every one interested in the pitot tube sooner or later wants to know what this suction is. The suction evidently depends for one thing on the shape of the orifice and the tip.

83 In No. 18 the pressure due to velocity may be found from the velocity of tip less the air motion

$$\begin{array}{lr} \text{Tip velocity} = \dfrac{41.67\text{ in.} \times \pi\, 2 \times 507}{12} = & 11080 \\ \text{Air motion in path and moving with tip} = & \underline{245} \\ \text{Velocity of impingement on tip} \quad = & 10835 \end{array}$$

From

$$\text{velocity, ft.} = 1097\sqrt{\frac{\text{velocity pressure}}{\text{weight 1 cu. ft. air}}}$$

$$\left(\frac{10835}{4085}\right)^2 = 7.02\text{ in.}$$

which is the velocity pressure due to the relative velocity of the tip to the air in inches of water. The centrifugal heads in column *B* were computed by logarithms by the method in Par. 74 except that the final head in inches of mercury was multiplied by 13.61 by slide rule.

84 For computation of centrifugal head for the static runs pressure at the axis is $B\alpha - k$, if the pressure in the tip is atmospheric, and the mean density in the arm is slightly less than for the impact runs, since for the impact runs pressure at the axis is very nearly atmospheric.

85 It can be shown that the pressure due to impact on the tip, Fig. 2, would exactly equal the centrifugal head if the fluid in the arm were of uniform density.

86 *Static.* Static runs Nos. 19 to 33 are with good forms of static orifice. It is thought that the two sets of runs 22 to 26 and 32 and 33 deserve to carry greater weight than the others. However, the mean of all for pressure in tip divided by the centrifugal head is 0.00625. It is evident by comparing the first part of these runs with the last that the holes are small enough so that the static does not depend on the inner tube in any way.

87 The static orifices are simply holes 0.02 in. in diameter drilled through the tubing which was about 0.03 in. thick. The holes were clean and the outer edges square, the little burr being carefully filed off.

88 Runs Nos. 34 to 37 are with slotted static orifice. The outer tube was about 0.4. in. in diameter, its inside diameter being about 0.28 in. With no inner tube the results are poor even with this thick tube. With an inner tube central and fitting the outer within about 0.015 in. the result was fairly good, but when the inner tube was made 0.02 in. smaller in diameter the result was poor again. It is very interesting to know that when fine mesh wire cloth was wrapped around the tube outside the slots the results are good and agree with those from the small drilled holes. Just how accurate the gauze static would be was not found.

89 By comparison with standard static in air flowing in a pipe the results are found to be considerably worse with a thin outer tube than with a thick one using the same inner tube. An instrument with $2\frac{1}{4}$-in. slots $\frac{1}{16}$ in. wide in $\frac{3}{8}$ in. tubing $\frac{1}{32}$ in. thick and with $\frac{7}{32}$ in. inner tube gave static pressure readings 14 per cent of the velocity pressure too high.

90 By comparison in this way it was also found that holes $\frac{1}{16}$ in. in diameter in the $\frac{1}{4}$ in. regular tip which is $\frac{1}{32}$ in. thick give poor results.

THE PRESENT STATE OF DEVELOPMENT OF LARGE STEAM TURBINES

By A. G. Christie, Published in The Journal for May

ABSTRACT OF PAPER

This paper deals with the present state of development of leading types of large steam turbines, some details of construction, the commercial results obtained and some new uses to which steam turbines have been put. It also points out the probable tendencies of steam-turbine development.

For the purposes of discussion, large steam turbines are divided into two types, fundamental and modified or combined. The weak elements of the fundamental types are discussed and the advantages of the new types pointed out. Some new constructions are shown in section.

The details of construction are discussed fully. A table with brief notes on the details of construction of individual types is included for rapid comparison of the practice of various manufacturers. The different types are also compared on the basis of efficiency as shown by published results of tests.

The present status of low-pressure turbines, turbo-compressors, turbo-driven pumps, geared turbines and marine turbines is discussed briefly. The closing paragraphs deal with the probable trend of steam turbine development and future possibilities.

CLOSURE[1]

The Author. In the discussion it has been held that as high an efficiency ratio cannot be obtained with high vacuum and low superheat as with moderate vacuum and high superheat. This is probably true of the Curtis turbine, since it does not apparently utilize high vacuum as economically as other types, which was pointed out in the paper itself. This is therefore a characteristic only of this type of turbine.

If a turbine is designed for a large amount of available energy there is no reason why fully as high an efficiency ratio should not

[1] The discussion was published in The Journal for August.

Presented at the Spring Meeting, Cleveland, 1912, of The American Society of Mechanical Engineers, 29 West 39th Street, New York. All discussion is subject to revision.

be obtained as with a small amount of available energy. Sufficient data is not at hand to warrant a general statement for all types that such conditions as above referred to cannot be fulfilled.

In regard to the increasing use of the efficiency ratio as a measure of steam turbine economy, it is interesting to note that a leading European firm includes a statement in their steam turbine specifications to the effect that should the steam or vacuum conditions at the time of the tests vary from those specified, the efficiency ratio of the unit shall not exceed that represented by the guarantee.

The author recognizes that the efficiency ratio alone does not provide all the information desirable for a given turbine, but when used in conjunction with specified steam and vacuum conditions, it provides the most reliable index of the actual performance of the steam turbine.

It is very gratifying to note the attention given to the present uncertain state of steam turbine ratings. It is to be hoped that the Committee on Power Tests of the Society will bring in an early report on this subject and fix certain standards of rating.

Objections have been raised to the Erste Brünner tests. Before including these in Table 2, inquiry was made as to their reliability. These tests were conducted by Prof. K. Koerner, one of the most prominent of Austrian engineers, who substantiated the figures quoted.

Mr. H. Zoelly presented a paper[1] at the 1911 meeting in Zurich, Switzerland of the Institution of Mechanical Engineers in which he made the statement quoted in Par. 51. He gave his reasons for such an opinion and also presented results to verify his statements. An abstract of this paper was printed in Engineering, April 29, 1911, and can be referred to by those interested.

Mr. Crissey presents a very excellent classification of steam turbines. However, for the purpose of this paper, the author preferred to classify turbines not so much as pure theoretical types, but under a classification of names and types with which engineers in general are familiar, hoping thereby to avoid confusion.

There seems to be a diversity of opinion regarding blade erosion in Curtis turbines. Some users have reported serious erosion in much less time than Mr. Moyer quotes. These difficulties are being steadily overcome by improvements in design.

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[1] Steam-Turbines, Proc. I. Mech. Engrs., 1911, No. 3, p. 681.

Regarding Mr. Hodgkinson's statement concerning blade troubles, the author fails to recall in his own experience, under ordinary conditions of operation, a case of blade failure due to centrifugal force alone.

The value of test results supplied by manufacturers has been criticised. There have been cases in the past where such results have been questionable, but in general manufacturers now find it advisable to publish only such tests as represent the actual and usual performance of their machines. It is frequently the case that steam turbines sold subsequent to the publication of such tests must operate with the same degree of economy as represented by the test results.

The corrections for Table 1 given by Mr. Hodgkinson are very welcome, bringing the data presented strictly up to date. The description of the new type of Westinghouse blading is also of considerable interest to engineers.

In connection with Dr. Stodola's tests of a Sulzer turbine quoted by Mr. Moyer, it was the author's impression when visiting Sulzer's works that while their Diesel engines probably excelled all other makes, their steam turbines were not representative of the best advanced practice in Curtis-Parsons construction, owing to characteristics of general design and particularly to their Parsons blading. This blading was much heavier in section than the usual form, and was not shrouded or otherwise reinforced. The tests quoted amply bear out this impression, and the author believes that such results should not be considered as typically representative of the whole class of Curtis-Parsons turbines.

The test results presented by Mr. Emmet for Curtis turbines show decided improvements in design over the earlier Curtis turbines in Table 2, but still do not exceed the Curtis-Parsons or Curtis-Rateau results. Table 4 presents some results of tests which were available subsequent to the preparation of Table 2, and which show some remarkable performances of Zoelly turbines. The Bellis and Morcom tests were on some of their earlier machines, and it is reasonable to expect considerable improvement in units now under construction. The results obtained by the Ljungström turbines are remarkable for high efficiency.

TABLE 4 ECONOMY TESTS OF HIGH PRESSURE STEAM TURBINES

EFFICIENCY RATIOS BASED ON E. H. P.

Maker of Turbine	Type	Date of Test	Load, Kw.	R.P.M.	Steam Pressure, Lb. Absolute	Temperature at Throttle, Deg. Fahr.	Vacuum Referred to 29.92 In. Barometer	Condenser Pressure, Lb. Absolute	Lb. of Steam per Kw-Hr.	B.t.u. per Kw-Hr.	Heat Utilized per Lb. of Steam	Heat Available per Lb. of Steam	Efficiency Ratio	Reference
Escher, Wyss & Co.	Zoelly	1912	1275	3000	196.3	588	28.99	0.455	12.59	15940	268.6	418.1	64.3	Zeit. d. v. d. Ing., 4 13 '12
Escher, Wyss & Co.	Zoelly	1912	5418	1500	180.7	588	28.74	0.580	11.95	15060	285.5	402.1	71.0	
Elsassische Maschinenbau Gesellschaft	Zoelly	1912	10006	1250	167.8	565	28.65	0.625	12.01	14975	284.2	389.1	73.0	
Belliss & Morcom	Drum Impulse	1911	3000	1500	192.5	558	27.92	0.981	15.0	18420	227.3	372.7	61.0	Manufacturers' Data
Belliss & Morcom	Drum Impulse	1911	2500	1500	164.7	466	26.92	1.471	16.95	19815	201.2	323.6	62.2	
Aktiebolaget Ljungström Angturbin	Radial Reaction (Ljungström)	1911	1000	3000	181.7	669	28.55	0.671	11.68	15130	292.0	414.9	70.4	Engineering, 4 19 '12
		1911	972	3000	176.7	667	28.58	0.657	11.55	14960	295.3	414.2	71.3	

Marks and Davis tables used.

The design of the details of this turbine shows a high degree of originality on the part of the inventor. The turbine is said to be very cheap to build, but before putting it on the American market many parts would have to be simplified and improved, particularly the bearings, steam packing and blade holding rings.

BITUMINOUS COAL PRODUCERS FOR POWER

By C. M. Garland, Published in The Journal for June

ABSTRACT OF PAPER

The paper describes the apparatus and general arrangement of bituminous coal producers as designed for power. The scrubbing apparatus is described in detail, data given on its efficiency, and the amount of solid matter delivered in the gas. Data on the efficiency of the plant, composition of the gas, and operating costs, together with brief discussions on these items are also included. Figures for the first cost and operating costs at full load for a 1200-b.h.p. plant are given in such a way as to make them applicable to different conditions of fuel and load.

CLOSURE [1]

The Author. Regarding the question of the low grade fuels high in ash, brought up by Mr. Hood, the writer would state that in this country at the present time there are producers operating on fuels containing as high as 22 to 23 per cent of ash. These are principally anthracite fuels and the producers that have proved successful in using these fuels are of the rotating table type, in which an ash bar rakes the ash from the table, dropping it into a water seal or a cone bottom. It is quite probable that this same type of producer will use fuels successfully that contain ash up to 30 and 40 per cent.

The CH_4 as given in the paper in the different analyses, was carefully determined and the writer feels that the results are accurate for two reasons. First, the calorific value of the gas, as calculated from the analyses, checked within about 1 per cent of the calorific value of the fuel as carefully determined by a Junkers calorimeter. This would indicate that no serious error was made in the determination of the CH_4. The carbon monoxide

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[1] The discussion was published in The Journal for August.

Presented at the Spring Meeting, Cleveland, 1912, of The American Society of Mechanical Engineers, 29 West 39th Street, New York. All discussion is subject to revision.

was absorbed in three successive pipettes, so that there is every reason to believe that essentially the last trace of this constituent was removed. The writer has conducted a number of investigations on the accuracy of the removal of the carbon monoxide from producer gas and finds that if reasonable care is exercised in the analysis there is no danger whatever of having this constituent appear in the determination of the CH_4. To insure this, however, three pipettes must be used and the solution in the third pipette maintained fresh.

Referring to Mr. Marsh's remarks regarding the question of the use of producers for peak loads of 10,000 kw. and over, and also to Mr. Dreyfus' remarks along the same lines, the writer can only reply that under certain conditions a producer plant is economical and desirable for such loads. Under ordinary conditions, however, the large central station in cities is operating under a very low load factor, probably under 50 per cent, and due to this low load factor the straight producer plant cannot compete with steam turbines. Where the load factor, however, is 75 per cent or over, the producer plant in practically every case would be more economical than a steam plant. For plants of 1000 kw., it can be said at the present time there are a number of such installations in operation that are producing highly satisfactory results. For plants of this size and with a load factor not under 50 per cent the producer plant will show a decided saving over a steam plant. The writer feels that Mr. Dreyfus is in error when he states that with the prevailing price of fuel a producer is commercially limited to sizes under 1000 kw.

Mr. Baker's statement that the temperature of 212 deg. is sufficient to prevent corrosion in exhaust fire boilers is correct, providing the load on the engine is uniform so that in the case of load fluctuation the temperature of the gas will not be decreased below this point. In actual practice, however, there is always more or less fluctuation in the load, so that the heating surface in the boiler should be so designed that the minimum temperature should be 212 deg. This will, of course, mean that the maximum temperature of the gas leaving the boiler will be much higher than this, anywhere from 220 to 300 deg. fahr. It is, therefore, this practical consideration that induces the writer to state that the gas should not be reduced below 220 deg. fahr. The writer feels that the piston cited by Mr. Baker is a very unusual case, for it would hardly be possible to honeycomb a block of steel in diluted sulphuric acid within 24 hours time, unless there were very unusual conditions of composition existing in the steel. The fact that the steel was honeycombed would seem to indicate that the steel was not of uniform composition. The acidity caused from the presence of sulphur in exhaust gases, so far as the writer's experience extends, does not produce serious corrosion on cast-iron parts.

FOREIGN REVIEW

BRIEF ABSTRACTS OF CURRENT ARTICLES IN FOREIGN PERIODICALS

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FOREIGN REVIEW

The aim of the Foreign Review is to present, within the available space, the main data contained in the articles indexed. Where possible, reference is made to English or American publications containing fuller information on the subject treated. Measures are given both in original units and their English equivalents. In many instances, engravings and tables are reproduced. Opinions expressed are those of the reviewer, not of the Society. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *t* theoretical. Articles of exceptional merit are rated *A* by the reviewer.

Aeronautics

COMPARISON OF EXPERIMENTS WITH AN AIRSHIP AND MEASUREMENTS MADE ON ITS MODEL (*Vergleich von Versuchen auf einem Flugzeug mit Messungen an dessen Modell*, C. Wieselsberger. *Zeits. für Flugtechnik und Motorluftschiffahrt*, vol. 3, no. 13, p. 174, July 13, 1912. 2 pp. 3 figs. *c*). Account of a paper published by G. Eiffel in the "*Inauguration du nouveau laboratoire aérodynamique*" on the comparison of data obtained from actual measurement on an airship specially fitted with proper measuring appliances and data obtained in the laboratory on a model of the same airship placed in an artificial air current. This question is very important at the present time in view of the installation of numerous aerodynamic laboratories in various parts of the world. It was found that the values obtained in the laboratory are surprisingly close to those obtained in actual flights, the two, on the average of seven measurements, being within 1 to 2 per cent of each other, which is close enough for all practical purposes, and proves that laws of flight may be profitably investigated on models in laboratories. According to the law of similarity, a model tested at the same velocity as an airship of working size could not be expected to give absolutely the same results, since this is possible only if the respective velocities are inversely proportional to the length of the machines, which cannot always be done conveniently because with high velocities in actual flight and models of moderate size the velocities of air in the laboratory work might approach to that of sound, and then, according to Blasius, the elasticity of air could no longer be neglected.

The article contains also a description of a simple application of the

Venturi tube for measuring the velocity of an airship with respect to the air. The air flows through a nozzle (Fig. 1) consisting of two truncated cones, with a U-shaped manometer tube attached at *A* and *B* as shown. The pressure difference *p* is proportional to the square of the velocity of the airship with respect to the air. The constant of the instrument is determined by calibration. The position of the meniscus may be recorded by photography.

Air Machinery

Free Air or Circulating Air? (*Frischluft oder Zirkulationsluft?* Oscar Gerold. *Dinglers polytechnisches Journal*, vol. 327, no. 29-31, July 20-August 3, 1912, 7 pp. *p*). Discussion of *ventilating problems in textile plants.* Exhaust fans in textile plants have the disadvantage of carrying away not only air containing dust and lint, but also the so-called false air, i. e. comparatively clean air, the handling of which involves unnecessary loss of power, besides other disadvantages. A better system therefore appeared to be to place suction apparatus at the dust-producing machinery so as to keep the lint out of the work rooms. The air carried away by these apparatus was however too dirty to be let free. It could be cleaned either in rotary strainers or in filters, the former delivering air if not absolutely pure, at least clean enough to be used without too bad consequences, while the filters delivered practically pure air. The exhaust fans of the suction apparatus carry however much more air than is really necessary to convey the dust, and while for hygienic reasons change of air three times per hour is quite sufficient, the action of the exhaust apparatus leads to the air being changed in the room sometimes as often as twenty times per hour. As a result immense amounts of properly conditioned, i. e. humidified and heated air are uselessly blown into the atmosphere, and have to be replaced by new conditioned air at a certain expense. When filters are used, the air is so pure that it may be returned to the workshop, with an addition of about 15 per cent new air, this amount being sufficient to provide for the three times per hour renovation of the air. The question as to which system to use, free air with rotary strainer (*Zyk-*

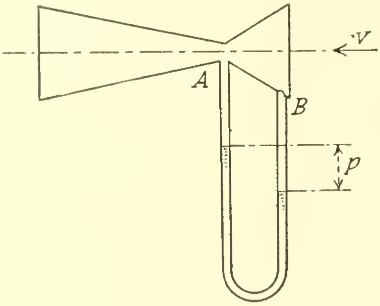

Fig. 1 Venturi Tube for Measuring the Velocity of an Airship with respect to the Air

lonen) or air of circulation with filters, is said to be a burning one in Germany, the main argument of the manufacturers of free air apparatus being that air returned to the workshop from the filters is polluted by dust and carbon dioxide. The investigation of Dr. Götting, court chemist in Breslau, Germany, has shown however that filtered air does not practically differ from fresh air with respect to the contents of dust and carbon dioxide. The author proceeds therefore to compare the cost of installation and operation of both systems in Germany, and shows that with free air, as compared with air of circulation systems, the initial cost of this installation is about 40 per cent, and the cost of operation over 50 per cent higher.

Firing and Fuels

NEW METHOD OF FIRING WITH LIGNITE AND LIGNITE BRIQUETTES (*Ein neues Verfahren zur Verfeuerung von Braunkohlen und Braunkohlenbriketts*, Pradel. *Braunkohle*, vol. 9, no. 14, p. 209, July 5, 1912. 1½ pp., 2 figs. *d*). Description of a new furnace arrangement patented by Ernst Lochner, of Jena, Germany, for firing with lignite and lignite briquettes. It is a self-feeding furnace but so arranged that the front part of the grate is fully exposed to view, while the clinker grate is set in the coldest part of the furnace. To make this possible, the grate is inclined not from the front part towards the bridge as usual, but in the opposite way, so that its highest part which receives the fuel is located just under the fire bridge, and its lowest part, the clinker grate, at the fire door. These furnaces have been in actual service for several months, and are said to work satisfactorily, and to be especially good on the arched parts of the furnace.

Hydraulics

FUNDAMENTAL THEOREMS OF HYDRAULICS OF RIVERS (*Théorèmes fondamentaux d'hydraulique fluviale*, A. Merten. *Annales de l'Association des ingénieurs sortis des écoles spéciales de Gand*, vol. 5, no. 2, 1912, p. 207. 5 pp., 1 fig. *t*). The theorems of hydraulics relating to the flow of water in open channels are derived from an investigation of the flow of only prismatic bodies of water (canals), and are applied to the whole field by the use of a postulate claiming that it is legitimate to apply to the *flow of natural bodies of water* theorems deduced for prismatic bodies of water. The author claims to have formulated a theory by which the modifications produced in the axes by the variations of the volumes of flow may be directly considered. The differential equation of the axes is:

$$ids - dh \sqrt{1 - i^2} = \frac{q^2}{g} \cdot \frac{d\omega}{\omega^3} + \frac{q^2}{\omega^3} bxds$$

or

$$i - \frac{dh}{ds} \sqrt{1 - i^2} = \frac{q^2}{\omega^3} \left[bx - \frac{1}{g} \left(\frac{\delta\omega}{\delta s} + \frac{\delta\omega}{\delta h} \cdot \frac{dh}{ds} \right) \right]$$

or

$$i - \frac{dh}{ds} \sqrt{1 - i^2} = \frac{q^2}{\omega^3} \left[bx - \frac{1}{g} \left(\frac{\delta\omega}{\delta s} + l \frac{dh}{ds} \right) \right]$$

where i is the slope, s the distance along the axis from some origin, h the elevation of the water surface, q volume of flow, ω wetted area, x wetted perimeter, b coefficient of resistance. The lines in Fig. 2 are drawn with $\frac{dh}{ds}$ as abscissae, and the values of the first and second members as ordinates. Line I (first member) the point of intersection with the axis of the abscissae, is independent of the volume of flow, elevation of water surface, and everything except the slope $\frac{dh}{ds} = \frac{i}{\sqrt{1-i^2}}$. For the line II (second member) the shape of the section, elevation of water, volume of flow, and character of the river bed (roughness) are supposed to be given. The point of intersection at the axis of the abscissae is given by

$$bx - \frac{1}{g}\left(\frac{\delta\omega}{\delta s} + l\frac{dh}{ds}\right) = 0$$

or

$$\frac{dh}{ds} = gb\frac{x}{l} - \frac{1}{l}\frac{\delta\omega}{\delta s}$$

This point of intersection is independent of the volume of flow, but when the discharge increases, the ordinate of the origin increases also.

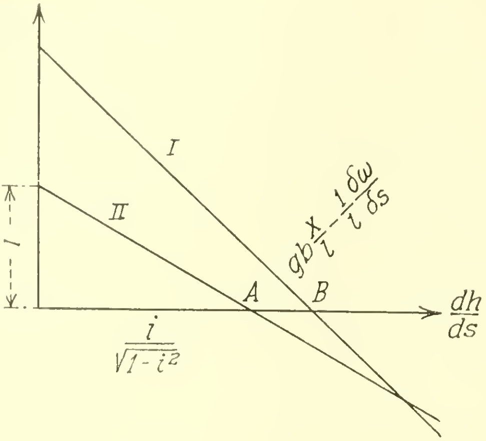

Fig. 2 Volume of Flow and Axis Diagram

Use of the diagram. For a given volume of flow line II may be drawn; the intersection of I and II has for its abscissa the inclination of the axis for given elevation of the water surface and volume of flow. There is however such a value of the volume of flow q^r for which the inclination of the axis is indefinite: this happens when I and II are parallel.

DISCUSSION OF THE DIAGRAM

First Case: $\frac{i}{\sqrt{1-i^2}} < gb\frac{x}{l} - \frac{1}{l}.\frac{\delta\omega}{\delta s}$

When q increases, $\frac{dh}{ds}$ decreases, except in the case of the sudden jump from $-\infty$ to $+\infty$ for q_r.

Second Case: $\frac{i}{\sqrt{1-i^2}} = gb\frac{x}{l} - \frac{1}{l}.\frac{\delta\omega}{\delta s}$

or $\frac{dh}{ds} = \frac{i}{\sqrt{1-i^2}}$: horizontality, independent of the volume of flow.

Third Case: $\frac{i}{\sqrt{1-i^2}} > gb\frac{x}{l} - \frac{1}{l}.\frac{\delta\omega}{ds}$

When q increases, $\frac{dh}{ds}$ increases also, except in the case of a sudden jump from $+$ to $-\infty$ for q_r.

The regions of

$$\frac{i}{\sqrt{1-i^2}} > gb\frac{x}{l} - \frac{1}{l}.\frac{\delta\omega}{ds} \dots\dots\dots\dots \text{(positive region)}$$

and

$$\frac{i}{\sqrt{1-i^2}} < gb\frac{x}{l} - \frac{1}{l}.\frac{\delta\omega}{\delta s} \dots\dots\dots\dots \text{(negative region)}$$

may be considered to be separated by *equality curves*, and as a particular case only one region may exist, as is generally the case with prismatic channels.

The axes may for the purpose of this investigation be divided into continuous parts. Then (a) if an axis is in the positive region, $\frac{dh}{ds}$ increases when q increases, except in case of a passage from $+\infty$ to $-\infty$; (b) if an axis is in the negative region, the opposite happens; (c) when an axis intersects an equality curve, it is horizontal.

Corollaries: if two axes are in the same region: (a) they can intersect only in one point; (b) the axis of the larger volume of flow is above the other downstream from the point of intersection, and below it upstream, if the region is positive, and vice versa. If two axes are located in two consecutive regions, they can have not more than two points of intersection; if they have a common point on the equality curve, they are tangent to it and can have no other common point.

Internal Combustion Engines

UTILIZATION OF EXHAUST HEAT FROM INTERNAL-COMBUSTION ENGINES (*Abwärmeverwertung bei Verbrennungskraftmaschinen*, K. Kutzbach, *Stahl und Eisen*, vol. 33, no. 28, p. 1133, July 11, 1912. 2½ pp., 6 figs. *d*). Discussion of the problem of exhaust heat utilization in connection with internal-combustion engines. This problem is more complicated and less attractive than the similar problem with steam engines. Comparatively a much smaller proportion of heat can be saved in the first instance (Fig. 3). The heat carried off by cooling water can be saved only partly in extremely rare cases, especially in large gas engines, because if the temperature of the water exceeds 40 deg. cent. (104 deg. fahr.) there is danger of pre-ignition. The exhaust gases cannot be used directly, like steam, because with their large volume they have a comparativly low heat content and give up their heat about 200 times slower than steam, and they contain as a rule various impurities, such as oil, sulphuric acid, and soot. Their only uses therefore appear to be for the production of steam for heating purposes, power production, and for gas producers. The author points out that where gas producers use fuels free of tar the waste heat may be utilized for making all the steam which they require, but when the fuel contains tar this cannot be done because tar rapidly destroys the boilers. The article contains a description of the horizontal and vertical types of boilers for utilizing the exhaust heat of internal-combustion engines constructed by the Maschinenfabrik Augsburg-Nuremberg. Vertical boilers are generally used for the production of hot water, horizontal for steam making.

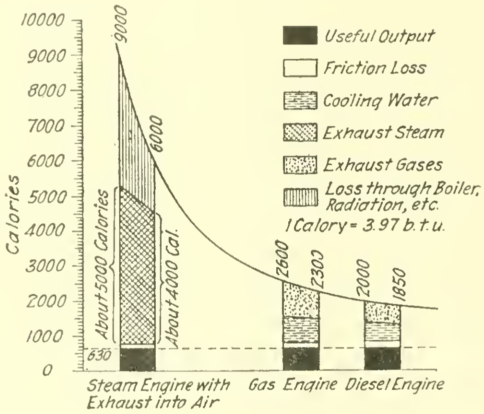

Fig. 3 Heat Balances of Steam, Gas and Diesel Engines

Efficiency of the Explosion Gas Turbine (*Der Wirkungsgrad der Explosions-Gasturbine*, H. Holzwarth, *Zeits. des Vereines deutscher Ingenieure*, vol. 56, nos. 24 and 25, June 15 and 22, 1912. 8 pp., 5 figs. *ec*). An article of controversial nature in which the author attempts to prove that his formulae and data are on the whole preferable to those of Professor Stodola as far as the calculation of the efficiency of the explosion gas turbine is concerned. He shows further that the heat processes in the gas turbine can be investigated only by graphical methods if heat losses and variations in the specific heats are to be taken into consideration: analytical methods are said to be entirely inapplicable for this purpose. The article also contains valuable data of tests on gas turbines made by the author since the publication of his book. The same journal, No. 25, June 22, contains an answer by Professor Stodola in which he defends his views and attempts to show that the gas engine cannot absolutely have as high an efficiency (30 per cent) as Mr. Holzwarth claimed for it. For the gas entropy table discussed in an appendix to Professor Stodola's article see p. 1397.

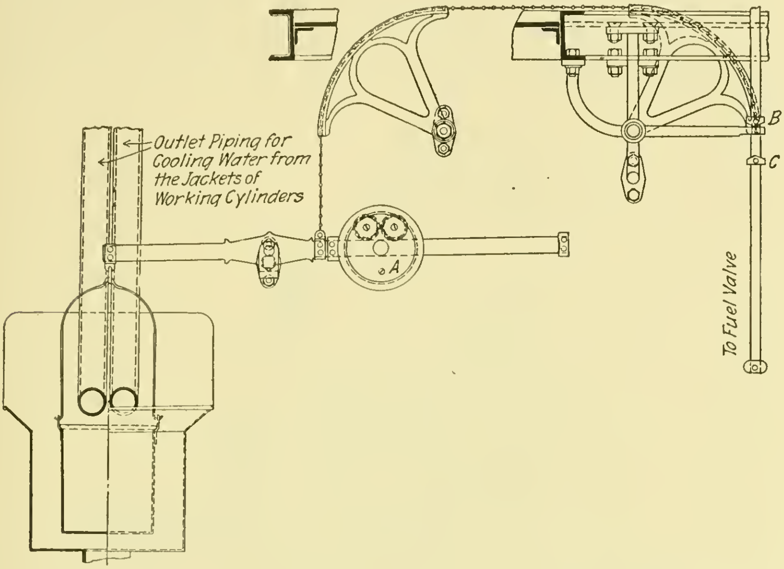
Outlet Piping for Cooling Water from the Jackets of Working Cylinders
B
C
A
To Fuel Valve

Fig. 4 J. A. Stroganoff Device for Stopping an Internal-Combustion Engine when Cooling Water Ceases to Circulate

Automatic Device for Stopping Internal Combustion Engines in Case of Lack of Cooling Water. Designed by J. A. Stroganoff of Twer, Russia (*Selbsstätige Anhaltsvorrichtung für Verbrennungsmaschinen bei Kühlwassermangel, Bauart J. A. Stroganoff* (Twer), N. Bikoff and G. v. Doepp. *Die Gasmotorentechnik*, vol. 12, no. 4, p. 62, July 1912. *d*). Description of a safety device intended to prevent overheating of an engine in case the supply of cooling water is cut off without being noticed by the engineer. A valve in the fuel piping of the motor is connected with one arm of a lever having on the other arm a vessel into which flows water from the cylinder jackets. When everything is in order, part of the water flows over the edge of the vessel while the rest escapes through little openings in the bottom of the vessel which, being full, occupies the lowest position possible with the prolongation *B* in Fig. 4, but when the flow of water ceases, the vessel empties, and the counterweight *A* moves up as far as *C* permits it, causing the valve in the fuel piping to be closed and the admission of fuel to the motor to be shut off.

Albaret Gasolene Motor for Farm Work (*Moteur à essence Albaret pour l'agriculture. Le Génie Civil*, vol. 61, no. 11, p. 226, July 13, 1912. 1 p., 3 figs. *d*). Description of the Albaret farm gasolene motor built by the Société des Anciens Établissements Albaret, of Rantigny, Oise, France. As shown in Fig. 5, the cylinder C and its water jacket are cast in one piece with the crank case C' provided with the opening P' for the inspection of the inside. By means of a connecting rod b, the piston P operates a handle m located at one of the extremities of the shaft a carrying at its other end the pulley-flywheel V; this shaft rotates inside a long and strong bearing supported by the crank case C', the flywheel V being given a curved shape in order to bring the plane of the center of gravity inside the bearing. Inside the crank case C' is placed the fuel tank R, the gasolene passing through the pipe t and pump p to the carburetor q, any excess of the fuel returning to the tank by the overflow pipe t'. There is no special lubricating apparatus, a suitable quantity of oil being simply poured into the crank case where the handle m is placed, with the level indicated by a tube outside. A high tension magneto takes care of the ignition, the timing being automatically regulated.

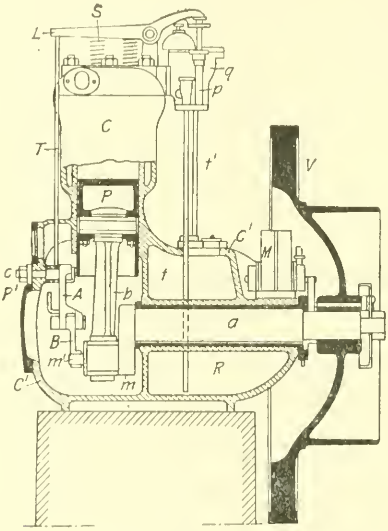

Fig. 5 Albaret Gasolene Farm Motor

The article describes also the system of governing which is of the hit-and-miss type, but rather complicated.

German Automobile Design (*Deutsche Automobilkonstruktionen*, *Auto-Technik*, vol. 1, no. 13, appended to the Allgemeine Automobil-Zeitung, vol. 3, no. 29, p. 33, July 19, 1912. 6 pp., 9 figs, *d*). Description of the Sperber 4-cylinder 18-h.p. automobile. The Sperber carburetor is shown in Fig. 6. The gasolene enters through 1, and is regulated by a float spindle in the usual manner. From the float chamber it passes through duct 2 into a ring-shaped groove in the female cone of clutch 3 provided along its inner circumference with fine nozzle-shaped openings; the gasolene squirts through these openings and forms an explosive mixture with the air coming from below. The mixture is regulated by the vertical displacement of the cone 4 which is free to move on the spindle of the throttle valve 5, and can be raised both by hand and by suction. The throttle valve may be regulated either from the accelerator pedal by a regulating rod, or by turning the bell-shaped case 6 from a lever on the steering wheel. The casing 6 is provided with spiral grooves in which run little screws 7 set in the throttle valve, and when the casing turns, the screws force the valve to move up and down; when the opening is throttled entirely, the valve goes down into the cylindrical guide 8. The air ratio of the mixture is regulated by the rotary slide-valve 9 varying the opening 10

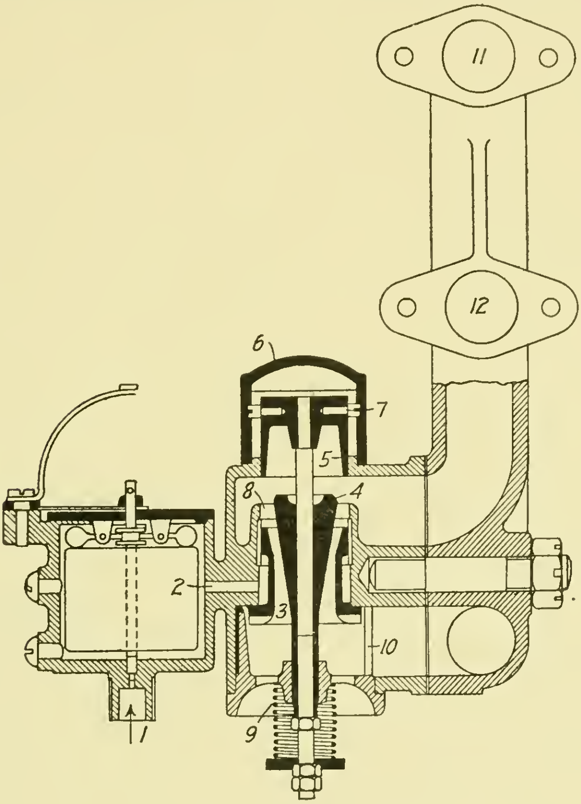

Fig. 6 Sperber Carburetor

for the admission of air (preheated). Owing to automatic air regulation, the carburetor, with the throttle valve open adapts itself to the speed of the motor, and delivers a correct mixture as well at high speed as at low. The bend connecting the pipe to the right of the carburetor contains two pipes leading in opposite directions: the first for warm air goes from 11 at the top down to the lower part of the carburetor, and passes on the way through some of the hot water around the cylinder casting, so as to preheat the air. The other flange 12 is screwed on to the suction piping. The whole suction piping is cast in one piece with the cylinder casting.

Investigation of the Working Process of a Two-Cycle Motor (*Untersuchung des Arbeitsprozesses eines Zweitaktmotors.* H. Scheit and Assistant Bobeth. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 22, p. 862, June 1, 1912. 8 pp., 18 figs. *cA*). Account of tests of a two-cycle motor made at the Royal Saxonian Mechanical and Technical Laboratory in Dresden, with a view of determining the working process of two-cycle motors. The motor tested was a single-cylinder motor-boat engine of the Ferro Machine and Foundry Co., of Cleveland, O., rated at 4 h.p., 800 r.p.m., 3¾ in. bore, 3½ in. stroke, jump spark ignition. Gasolene was used as fuel with an average upper heating value of 11054 WE (19,945 b.t.u. per lb.), and lower heating value 10334 WE, or 18,660 b.t.u. per lb. From the chemical analysis of the fuel it was found that 3.483 kg of

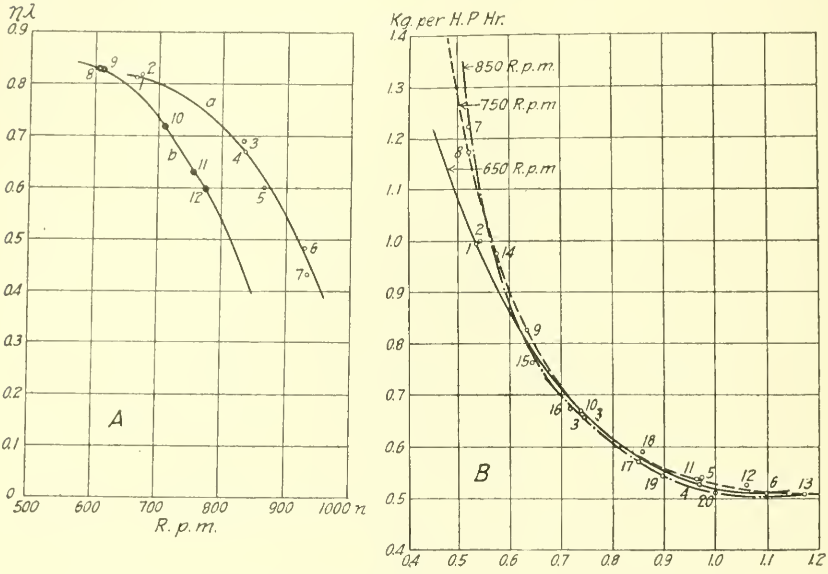

Fig. 7A Efficiency of Pump and Speed of Two-Cycle Motor
Fig. 7B Ratio of Mixture and Fuel Consumption of Two-Cycle Motor

oxygen were required for the combustion of 1 kg of fuel, this amount corresponding to 14, 91 kg of air, or at one atmosphere pressure and 15 deg. cent. (59 deg. fahr.) temperature 12.55 cbm (200 cu. ft. of air per lb. of combustible).

The first series of tests had the purpose of determining the relation between the efficiency of the fuel pump and speed of rotation of the motor. To reduce the resistance in the suction piping the throttling valve at the carburetor was thrown wide open, and the most favorable opening of the fuel nozzle and position of the spark determined for each speed by previous tests. The tests as shown in curve *a* (Fig. 7 A) prove that the efficiency of the pump at slow speeds is pretty high, but rapidly falls with the increase in the number of revolutions; this may be due either to the increase of the resistance in the suction piping as the velocity of air increases, or to oscillations in the sucked-in column of air. The fact that the inlet opening is uncovered only at the end of the stroke and that the length of time while it is uncovered decreases as the speed of the motor increases may be of importance. A motor with the inlet opening governed not by the piston, but by a valve in such a manner as to be open during the whole suction stroke would therefore be more efficient, provided it were simple in construction.

Curve *b* shows the efficiency of the pump as a function of the speed of the motor when the gasolene was shut off, and the motor driven by an electric motor worked only as a pump. The purpose of this test was to eliminate the influence of the resistance *s* in the exhaust and at the entrance of the charge into the working cylinder. The comparison of the two curves shows that in the second case with a constant back pressure equal to atmospheric, the efficiency decreases more rapidly with the increase in speed. It may be that during regular working of the motor the pressure in the cylinder at the end of expansion falls below atmospheric and the flow of exhaust gases produces a supplementary suction: this could not be definitely established by means of an indicator.

Influence of Ratio of Mixture on Gasolene Consumption. Three series of tests were made, with the motor working at about ¾ of its maximum output, and at 650, 750 and 850 r.p.m., with the indicated ratios of mixture, the best position of the spark for each ratio of mixture having been found by previous tests. The three lines (Fig. 7 B) show how substantially the fuel consumption rises with an insufficient supply of air. The lines for the three speeds of rotation nearly coincide, showing thereby that the speed of rotation has practically no influence on fuel consumption. A curve drawn by combining these three curves has shown that the most economic way of running the motor is with a 10 per cent excess of air.

Experiments were also made on the influence of air preheating, spark displacement, and cooling of exhaust gases. In general the tests have shown that the minimum fuel consumption of the Ferro two-cycle motor is 500 g (1.1 lb.) per h.p.-hour. This is more than the fuel consumption of a four-cycle motor of equal power which appears to be due partly to some fresh fuel being carried away by the exhaust gases and partly to a poorer ratio of mixture. The average effective piston pressure is also equal to only about one-half that in the four-cycle motor. The charging pump, though found to be very efficient absolutely, appears to be not strong enough in the case of a two-cycle motor to produce an effective scavenging of the cylinder, this task being made particularly hard by the fact that the time for it, which is only a fraction of the piston stroke, is very short.

The efforts of designers of two-cycle motors must therefore be directed towards improvement of the charging process. The most promising way appears to be the separate introduction of air and fuel, and the use of excess of air for scavenging.

Machine Shop

Double Milling Machine for Torpedo Screw Blades of J. E. Reinicker in Chemnitz-Gablenz (*Doppelte Fräsmaschine für Torpedo Schraubenflugel von J. E. Reinicker in Chemnitz-Gablenz*, F. Nickel, *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 23, p. 929, June 8, 1912. 3 pp., 6 figs. *d*). The machine mills the blade from a template, cutting in strips. The table carrying the screw has an automatic quick return and a stop motion which is thrown into gear upon completion of the work.

Broad or Narrow Welded Seams (*Breite oder schmale Schweissnahte. Autogene Metallbearbeitung in Acetylen*, vol. 15, no. 12, 1 p. *c*). The Technical Bureau of the Association for Autogenous Welding in reply to inquiries as to whether or not there is any technical advantage in making broad welded seams, meaning by this term melting the metal at a considerable distance on both sides of the welded joint, stated that the advantage was doubtful in most cases. Melting considerable amounts of metal in the immediate proximity of the seam and thoroughly joining it with the welded-in material helps to prevent oxidized particles from getting incrusted in the joint, but on the other hand subjection of the metal at the seam to prolonged action of the flame produces changes in its structure, and, owing to the high heat conductivity of metals, leads to considerable heat losses. A better way is to heat by a large burner the metal in the neighborhood of and parallel to the seam: this permits the heat to concentrate at the seam when a comparatively small burner is sufficient to make a good joint even in the case of thick plates, there being less possibility of overheating the metal than with the broad seam. In welding thick plates when the material along the seam is heated by a large burner to a dark cherry red color, the welding proper proceeds very rapidly without any need for melting large amounts of material around the seam.

Mechanics

Friction Gear with Variable Ratio of Speed (*Transmission par friction à rapport de vitesses variable*, P. Raes. *Annales de l'association des ingénieurs sortis des écoles spéciales de Gand*. Ser. V, vol. 5, no. 2, 1912. p. 188, 18 pp., 9 figs. *tdA*). Theoretical discussion of the principles of transmission by friction, particularly the problem of finding by calculus rolling surfaces of such a profile that by sliding along perpendicular axes they should produce simple rolling. The author shows that theoretically this condition is satisfied if one of the solids is generated by a parabola of the second degree rotating about its axis of symmetry, while the other solid is generated by a semi-parabola of a double parameter rotating about a tangent at the vertex. There are however considerable difficulties in embodying this principle in a variable-speed transmission in a small automobile, and further the author shows that the condition of simple rolling is not of sufficient importance to justify a very complicated system of transmission. The author further analyzes the action of a disk friction gear, such as is used in some actual automobile construction.

Materials of Construction

TESTS OF BOILER PLATES WITH LAP WELDED JOINTS (*Versuche mit überlappt geschweissten Kesselblechen*, B. Zwiauer, *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 22, p. 877, June 1, 1912, 4 pp., of which two are tables, 5 figs. *e*). Tests of lap welded joints, mainly with a view to the existing German regulation for the inspection of boilers.

Steam Engineering

ELECTROLYTIC OIL SEPARATION (*Elektrolytische Oelabscheidung*, W. Heym. *Elektrochemische Zeits.*, vol. 19, no. 4. p. 21, July 1912. 3 pp. *td*). Oil is carried away by the exhaust steam partly in the form of a finely divided liquid, partly in that of vapor, a considerable amount of oil or some of its constituents being vaporized no matter what is its boiling point. However well oil separators working on the mechanical principle be arranged, they can catch practically only the oil present in the form of a spray, about 75 per cent of the total, while the rest escapes and reaches the condenser. It is there transformed by the action of the water into minute spheroidal corpuscles, and is partly carried off with the feed-water, partly settles on the condenser walls, and helps to form a very unsoluble crust. The properties of this crust are determined by the fact that these oil corpuscles are charged with electricity. The crust can therefore be treated just like every other colloidal solution, and the oil particles may be settled down by being acted upon by a colloidal solution carrying an opposite charge, e. g., by colloidal iron hydroxide. On this property is based the *oil separating process* of Davis-Perret. It consists mainly in electrolysing the water of condensation by means of iron electrodes, there being added to the water some electrolyte, usually sodium carbonate, to make it more conductive. A basic salt is formed, and the alkalies settle at the anode, the oil particles being caught in this precipitate, and practically entirely separated out by subsequent filtration. The efficiency of the Davis-Perret oil separating process has been found to be as high as 98.5 per cent. It is further stated that since by this process a precipitate is formed which is not a pure hydroxide, but a basic salt, a certain excess of it reaches the boiler and makes the water alkaline; this in its turn makes the ability of the water to decompose iron lower than with pure water, owing to the fact that the ability of water to affect iron is proportional to the hydrogen content of the water. The article also describes how a simplified type of Richard's nepthelometer may be used for the determination of the contents of oil in water.

NEW PATENTS IN THE DOMAIN OF STEAM BOILER FIRING (*Neue Patente auf dem Gebiete der Dampfkesselfeuerung*, Pradel. *Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 35, no. 30, p. 315. 2 pp., 10 figs. *d*). Brief accounts of new German patents granted during the first half of 1912 for appliances in steam boiler firing, mainly grates.

ON THE EXPANSION OF SATURATED STEAM (*Sur la détente de la vapeur d'eau saturante*, A. Leduc. *Comptes rendus de l'Académie des Sciences*, vol. 155, no. 1, p. 33, July 1, 1912. 2 pp. *tA*). The object of the author's investigation was to find the precise value of n in Rankine's formula: pv^n = const. for the expansion of dry saturated steam on the basis of numerical data obtained by Holborn and Henning and Dieterici, and the author's own formula of state for real gases

$$Mpv = RT\phi$$

where M is the molecular mass of a gas, R* an absolute constant common to all gases; ϕ is called by the author the molecular volume of a gas, and is the ratio between the relative (say with respect to air) molecular weight of a gas to its relative density.

Let a unit of mass of the fluid pass through Rankine's cycle between T_0 and T_1, and let x be the quality of the steam at the end of adiabatic expansion. If k be the specific heat of water at T deg. under the pressure of saturated steam, and L its latent heat of evaporation, then

$$\int_{T_0}^{T_1} k\frac{dT}{T}+\frac{L_1}{T_1}=\frac{L_0}{T_0}x$$

A unit of mass of steam occupies a volume v_1 at T_1 under the maximum pressure p_1, while at T_0 it is only the mass x that occupies the volume v_0 under the maximum pressure p_0 (the volume $1 - x$ of the condensed water being negligible). Hence

$$M\ p_1\ v_1 = RT_1\phi_1$$

and

$$M\ p_0\ v_0 = xRT_0\phi_0$$

from which

$$\frac{x\ p_1v_1}{T_1\ \phi_1}=\frac{p_0v_0}{T_0\phi_0}$$

which combined with

$$p_1v_1^n = p_0v_0^n$$

gives

$$\frac{n-1}{n}=\frac{\log T_1-\log T_0+\log\phi_1-\log\phi_0-\log x}{\log p_1-\log p_0}$$

Table 1 has been calculated for intervals of 10 deg.

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* In a previous investigation (Annales de Chimie et de Physique, septième série, tome 15, p. 92) the author found that R in C.G.S. units, with the molecular mass of oxygen taken as 32, is equal to 8319×10^4, or in centimeters of mercury $R' = 6237.2$. He refers in a footnote for the notation in the present article to his former article in the *Comptes rendus*, vol. 153, p. 51, which does not contain it. The reader is referred to the above quoted article in the *Annales*.—EDITOR.

TABLE 1 PROPERTIES OF SATURATED STEAM FOR TEMPERATURES BETWEEN 160 AND 80 DEG. CENT. (320 TO 176 DEG. FAHR.)

Deg.	x	n	Deg.	x	n
from 160 to 150	0.9829	1.142	from 120 to 110	0.9810	1.142
from 150 to 140	0.9824	1.144	from 110 to 100	0.9805	1.140
from 140 to 130	0.9820	1.144	from 100 to 90	0.9800	1.137
from 130 to 120	0.9815	1.144	from 90 to 80	0.9794	1.134

The article contains also an additional table showing that slightly different results are obtained with the Regnault data.

The value of n passes through a maximum at about 135 deg. Nevertheless Rankine's formula may be used with n constant, provided its value be taken as 1.14 when the temperature of steam varies between 160 and 60 or even 40 deg. cent. (320 to 140 to 104 deg. fahr.), and as 1.135 when the engine works with a low pressure and has a very well cooled condenser. With the fall of initial temperature x decreases. From Table 1 it may be seen that x, after an expansion of θ deg. from the temperature of 160 deg. cent. may be exactly represented by

$$x = (0.9983)^{\theta}\left[1 - \frac{\theta(\theta-1)}{2} \times 5 \times 10^{-6}\right]$$

but in the case of steam engines practically for all initial temperatures the following simpler formula may be used:

$$x^1 = (0.9981)^{\theta}$$

Improvement of Steam Turbine Efficiency by the Use of "Exhaust Disks" (*L'amélioration du rendement des turbines à vapeur par l'emploi des "disques d'échappement,"* Delaporte. *Revue de mécanique*, vol. 30, no. 6, p. 517, June 30, 1912. 7 pp., 7 figs. *ed*). The last row of blades has always long blades as compared with the thickness of the distributed steam films, and the efflux comes out in an indefinite film, that is without the separate stream lines being given an opportunity to expand in the direction of the axis of a blade. Owing to the immediate proximity of other vapor films, the films are not allowed to expand in the perpendicular direction *ab* (Fig. 8, A) either. The guiding *ac* prevents entirely any change of the direction of the films in the triangular space *abc*: there can be therefore no expansion, and the pressure P at *ab* is constant for the entire space, and particularly for the section below disk *bc*. As a result, the difference between P and the pressure p at the plane of outlet of the disk and at the exhaust gives only to the steam an acceleration A parallel to the axis of the disk; the change in direction produced affords the films the possibility to thicken in accordance with the increase of the relative velocity of the steam from V to V', but as far as useful work is concerned this acceleration is entirely lost, because it is not accompanied by a change in the moment of the quantity of motion with respect to the axis. *There is therefore a certain limited pressure beyond which the turbine ceases to benefit by any further increase of expansion.* This limiting pressure P is characterized by the relative velocity at the outlet of the disk being 360 meters (1181 ft.) per second, the output being

$$\triangle = 0.97\,Ps$$

where $\triangle$ is in kg per hour, and s the total section open to the passage of the steam in sq. mm. (1 sq. mm. = 0.00155 sq. in.), Hence

$$P = 1.03\,\frac{\triangle}{s}$$

The fact that there are many correct graphical proofs of the important advantages derived by turbines from reducing the aval pressure does not conflict with the above statement: all such proofs hold only for cases where the ratio $\frac{\triangle}{s}$ is small and the pressure P is below the pressure p of the condenser. But with the increase in the size of turbines which for economic reasons has not been followed by a reduction of their speed, the output $\triangle$ has increased, while the section s could not be increased at the same rate owing to its being limited by the strength of the materials used. A disk of 750 mm (say 30 in.) in diameter, with blades 100 mm (3.9 in.) long, and a coefficient of distribution λ' from 0.6 to 0.65 ($\lambda' = \frac{ab}{bc}$, Fig. 8A) represents about the limit of what can be done at 3000 r.p.m.

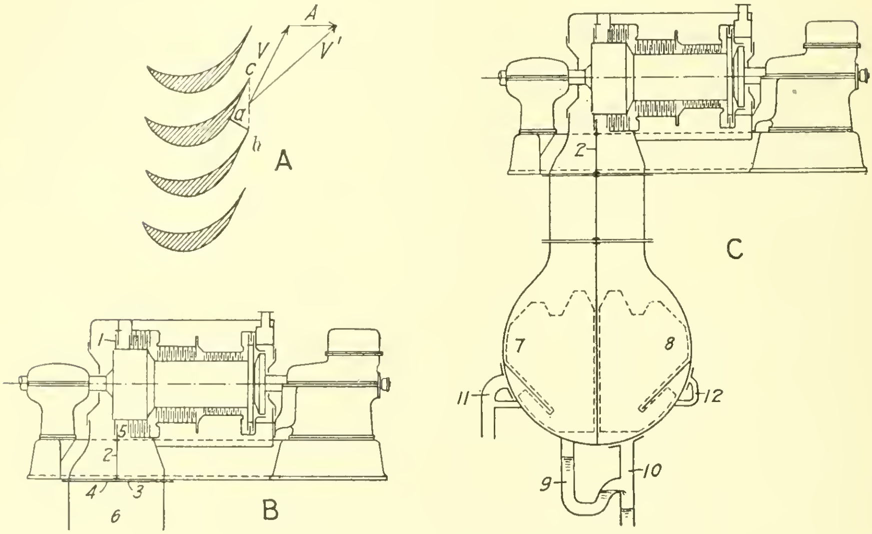

Fig. 8 Delaporte Exhaust Disks and Divided Condenser

without sacrificing the safety of operation, and that gives to s a maximum value of about 150,000. The ratio $\frac{\triangle}{s}$ thus becomes an element which can no longer be neglected.

As an example the author considers a turbine driving a 1500-kw. alternator at 3000 r.p.m. With average conditions of pressure, superheat and vacuum the steam consumption is 6.9 kg (15.18 lb.) per kw. per hour; hence $\triangle = 6.9 \times 1500 = 10{,}350$ kg, and

$$P = 1.03 \frac{\triangle}{s} = 1.03 \times \frac{10350}{150000} = 0.71 \text{ kg} = 1.009 \text{ lb.}$$

If the cooling condenser water be taken at 16 deg. cent. (60.8 deg. fahr.), and comes out of the condenser at 28 deg. cent. (82.4 deg. fahr.), the pressure p at the entrance to the condenser will be approximately that corresponding, according to steam table, to a temperature of about 5 deg. above the latter temperature, or $p = 0.051$ kg $= 0.729$ lb. The expansion from 0.071 to 0.051 kg is lost. But if a disk with $s = 150{,}000$ cannot take care of the entire steam at a pressure below 0.071 kg, another disk with equal section could drive off the steam at the pressure of 0.051 kg, if it had only to take care of a reduced amount of steam: this is the fundamental idea of the "exhaust disk." It is provided, like others, with a distributor, but this is placed between the two exhausts, and is traversed only by a part of the total flow of steam, the inlet connecting branch being contracted so as to limit the pressure in the main turbine to a value beyond which it ceases to be of benefit. Fig. 8 B gives an example of this arrangement: 1 is the exhaust disk of which the distributor seats on the cast-iron plate 2 between the branches 3 and 4, the first of which is contracted by a diaphragm (it is often more convenient to create a loss of pressure at 5). The two streams of steam reunite in 6 in the exhaust pipe. If five-eighths of the flow of steam be sent through the exhaust disk, 88 h.p., or about 3.9 per cent of the normal power, is recovered, which is significant considering the simplicity of the device. This gain can be still further improved by modifying the condenser arrangement somewhat. As shown in Fig. 8 C, the partition 2 is extended through the exhaust pipe right into the condenser which it divides into two parts. With regard to the circulation of water, the nests of tubes 7 and 8 are disposed in series with 7 receiving the fresh water. As a matter of fact this arrangement results in there being practically two separate condensers of which one, viz. that receiving the steam from the exhaust disk (the author calls it the cold condenser) permits of obtaining a final pressure p' considerably below the pressure p of the other condenser, or the one that would prevail throughout the entire condenser if it were not divided. Notwithstanding the difference of internal pressures, both condensers may be supplied with water by the same pump. With the modified condenser an increase in efficiency up to 6.8 per cent has been obtained in actual tests. The article contains full data of five tests.

Boiler Feed Regulator, Degrémont system (*Régulateur d'alimentation, système Degrémont*, F. D. *La Technique moderne*, June 15, 1912. ½ p., 1 fig. *d*). The Degrémont boiler feed regulator (Fig. 9), consists of a body A provided with a membrane *m*, and connected with the boiler by the pipes, *e. f. g.* and *h*, in such a way that the pipe *h* reaches the average level of water in the boiler. As long as *h* is in the water, the entire system of piping is filled by water, and equal pressures act on both sides of the membrane. But when the water in the boiler goes below the end of the pipe *h*, the water in this pipe runs out and the pipe is filled by steam. As a result, the pressure on the upper side of the membrane becomes predominant, owing to the additional pressure of the column of water in the pipe *g*, and the membrane is deflected downwards, and in doing so displaces by means of a suitably arranged lever the counterweight *b* acting on a flap valve *c* admitting feed water to the boiler. The admission of water continues until the water level rises above the lower end of the pipe *h*. This apparatus was shown at the Roubaix Exhibition of 1911.

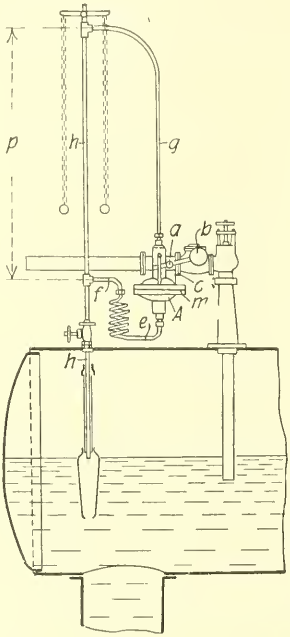

Fig. 9 Degremont Feedwater Regulator

Utilization of High Vacua in Turbines with Small Cross-sections of Outlet (*Die Ausnutzung hoher Luftleere in Dampfturbinen bei kleinen Austrittquerschnitten*, F. Lösel. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 25, June 22, 1912. 8 pp., 15 figs. *eA*). Description and data of an investigation undertaken to determine whether in turbines having cross-sections of outlet dimensioned for vacuum of 90 per cent higher vacua from 95 to 96 per cent may be utilized, and how the data obtained may be reduced to the basis of a vacuum obtained with a given temperature of cooling water of 25 deg. cent. (77 deg. fahr.: this was the temperature at which the nominal efficiency of the turbine tested was guaranteed by the manufacturers). The turbines were designed on the assumption that 90 per cent vacuum was reached without fail. The tests have shown: (*a*) that high vacua from 95 to 96 per cent may be fully utilized, and the output of the turbines grows with the vacuum; (*b*) that the thermodynamic efficiency referred to the state in front of the first diaphragm is the same for all vacua from 90 to 96 per cent, which shows further that the heat is well utilized, notwithstanding the small cross-section of outlet, even with the highest vacua. It is therefore quite permissible to accept the same efficiency in reducing the steam consumption to better vacua (the author states that this was denied by previous investigators); (*c*) the fact that the output of a turbine grows with the increase of the vacuum while the thermodynamic efficiency does not change may be explained only by the assumption that the velocity of the steam increases beyond the critical velocity, and that this high velocity is utilized. Since the pressure in the stage next to the last is the same for all vacua, it is only the last two stages that take part in the utilization of the high vacua. With a vacuum of 90 per cent for which the turbine was designed, already in the stage next to the last the pressure falls to that of the condenser, and the last stage not only produces no work, but consumes some through blade friction. The stage next to the last therefore has the best thermodynamic efficiency. The article contains complete data of these tests.

STEAM MACHINERY AND FIRE ENGINES OF LEONARD DA VINCI (*Dampfapparate und Feuermaschinen bei Leonardo da Vinci. Zeits. des Oesterr. Ingenieur und Architekten Vereines*, vol. 64, no. 28, p. 440, July 12, 1912. 2 pp. 7 figs. *h*). Description of a steam-gun and other steam machinery invented or described by Leonardo da Vinci, and particularly some experiments made by him to determine the volume and pressure of steam; also description of fire engines invented by him.

Thermodynamics

GAS ENTROPY TABLE FOR THE LATEST VALUES OF SPECIFIC HEATS (*Gasentropietafel für die neusten Werte der spezifischen Wärmen*, A. Stodola. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 25, p. 1008. 2 pp., 3 figs. *tA*.). The author bases his new table on the values of Pier as given in the German engineering pocket-book "Hütte" (in kilogram-molecules per deg. cent.):

Steam $C_v = 6.065 + 1.0 \times 10^{-3} t + 0.2 \times 10^{-9} t^3$
Carbon dioxide $C_v = 6.80 + 3.3 \times 10^{-3} t - 0.95 \times 10^{-6} t^2 + 0.1 \times 10^{-9} t^3$

which he replaces, for the purposes of the entropy table, by equations of the type (in kg.-molecules per deg. absolute):

$$C_v = a_v + bT + cT^2 \quad \ldots\ldots [1]$$

where $a_v = 4.67$, and b and c are so selected that the variations within the technically important range of temperatures, from 500 to 2000 deg. cent. absolute, be as small as possible. The above formulae become then

For steam $C_v = 4.67 + 1.263 \times 10^{-3} T + 1.18 \times 10^{-6} T^2$

For carbon dioxide $C_v = 4.67 + 8.004 \times 10^{-3} T - 2.337 \times 10^{-6} T^2$

which differ but very little from the values derived from the Pier equations, as shown by Fig. 10A. From C_v the specific heat for constant pressure may be obtained by the usual formula

$$C_p = AR + C_v \text{ or about } 2 + C_v \qquad [2]$$

The introduction of [1] and [2] into the formula of entropy in the case of gases gives

$$S = a_v \, lgn \left(\frac{T}{T_o}\right) + b(T - T_o) + \frac{c}{2}(T^2 - T_o^2) + AR \, lgn \left(\frac{V}{V_o}\right) =$$

$$a_p \, lgn \left(\frac{T}{T_o}\right) + b(T - T_o) + \frac{c}{2}(T^2 - T_o^2) + AR \, lgn \left(\frac{p}{p_o}\right)$$

which is the old Stodola formula of entropy for gases with the addition of the term containing c, and it is sufficient to plot on the line b in the entropy table the parabola $\frac{c}{2}(T^2 - T_o^2)$ from Fig. 10B, to obtain, within the range BB_2 the new entropy values. It is to be measured from the curvilinear axis c and gives as an adiabatic curve, i. e. with $S = \text{const.}$, the c parabola itself only displaced about BB_2, as indicated by the expansion line BC. This adiabatic curve preserves the property according to which the change of the heat content during a reversible change of state is equal to the area between the curve of state and the adiabatic curves down to the zero line.

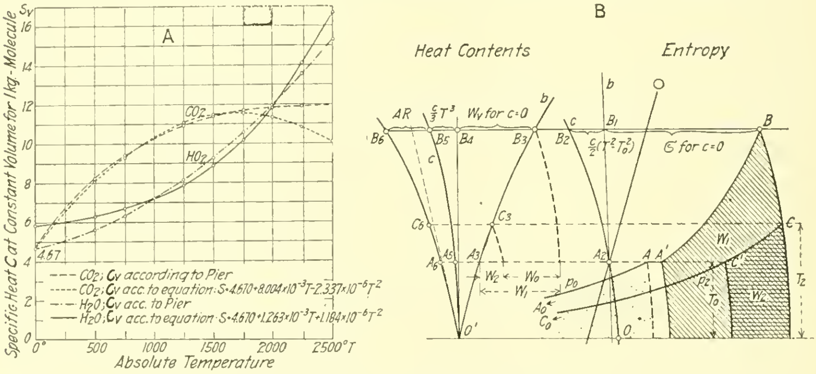

Fig. 10A Specific Heats of Carbon Dioxide and Steam

Fig. 10B Entropy Diagram for Specific Heats Varying according to Parabolic Law

The internal heat energy is expressed by

$$W_v = \int_o^T C_v dT = a_v T + \frac{b}{2} T^2 + \frac{c}{3} T^3$$

In the entropy diagram, B_3B_4 is for the heat energy W_v for $c = 0$; if the cubic parabola $\frac{c}{3} T^2$ be constructed from 0′, B_3B_5 will represent the total heat energy.

For the heat contents the author gives the expression

$$W_r = \int C_p dT = \int (C_v + AR) dT = W_v + ART$$

As an illustration the author constructs an entropy_diagram for the Holzwarth gas turbine.

Miscellanea

Motor Plows Testing (*Die Prüfung von Motorpflügen*, Dr. Martiny. *Der Motorwagen*, vol. 15, no. 20, July 20, 1912. *t*). The object of testing is the determination of the agricultural value of a motor plow. The agricultural value of plows with stationary cutters, such as the Stock and I.H.C. types, depends on the total cost of plowing a unit of ground (as compared with plowing by horses, steam, or electricity), cost of additional operations, and quality of work; that of plows with rotating cutters, such as the Köszegi type, depends also, perhaps primarily, on the influence of the peculiar handling of the earth, by this kind of plow: finely cutting and thoroughly mixing it. The total cost of plowing a unit of ground depends on the cost of upkeep of the machine (depending in its turn on the first cost, durability and strength of the plow), on the cost of fuel and attendance, and the amount of ground plowed per unit time to a given depth, apart from interruptions independent of the action of machinery. The output in units of plowed area cannot be determined just by measuring it, because it depends on the properties of the ground as well. (The article explains this in detail.) The efficiency of a motor plow depends further on the strain to which the plow is subjected at the tests. This may be done in two ways: either, while plowing, the gasolene consumption per hour, or number of revolutions per minute, and motor governing may be determined, or the turning moment of the engine may be measured by a torsion dynamometer; in laboratory tests the motor may be placed on the brake stand, and the hourly consumption of fuel, speed of rotation and governing may be determined.

In the case of motors with rotating cutters the most important are comparative tests of production. On the whole, tests of plows are not pure engine tests, but pass into the domain of plant life, in determining the agricultural result of the work delivered by the engine. This makes this kind of testing very difficult, complicated, and subject to errors which have not yet been quite elucidated: a statistical investigation parallel to the experimental appears therefore advisable.

Lubricants (*Analysen von Motor-Schmiermitteln*, N. Bikoff and G. v. Doepp. *Die Gasmotorentechnik*, vol. 12, no. 4, p. 63. *e*). Several tables containing data of tests of various lubricants for internal-combustion engines made at the International Exhibition of Internal Combustion Engines in St. Petersburg. Most of the oils tested were of Russian manufacture.

Patent Laws of Holland (*Das holländische Patentrecht*, R. Bauch. *Zeits. fur Elektrotechnik und Maschinenbau*, vol. 15, no. 29, July 17, 1912. 2 pp. *g*). The first patent law went into force in Holland on June 1, 1912, and requires the patent application to be presented in five copies. The patent office considers only whether the description and drawings are such as to make the invention clear to men familiar with the respective art. The question of novelty is not considered, but due notice of the application is given, and the application itself may be seen in Holland and its three great colonies by anybody interested who may subsequently make objections to the grant of the patent. A term of six months is set for objections, and a small fee must be paid, the purpose of the fee being to prevent people from making groundless objections. The application must be made by the inventor himself or his attorneys. If the invention is not worked during five years the patent may be cancelled by the patent office unless there is a good excuse for the default. Three years after the grant native industrial concerns may obtain a compulsory license to manufacture the patented article. Patent decisions may be appealed from to the general courts. Patent fees are: application fee 25 fl., fees for the patent, second and third years 50 fl. each, and an increasing fee for each subsequent year (a florin is equal to 40.2 cents).

Supplementary References

MOTOR FIRE ENGINES (*Foreign Review*, August 1912, p. 1241). Complete translation in *Automobile*, August 1 1912.

USE OF PREHEATED AIR IN INTERNAL-COMBUSTION ENGINES (*Foreign Review*, April, 1912, p. 615). Translation of the article from the *Gasmotorentechnik*, but without the diagram reproduced in the Foreign Review, in *Automobile*, July 18 1912.

GAS POWER SECTION

PRELIMINARY REPORT OF LITERATURE COMMITTEE

(XX)

ARTICLES IN PERIODICALS[1]

AIR TRANSMISSION FOR A GAS-ENGINED VESSEL, “CLOSED CIRCUIT,” *The Engineer* (London), July 26, 1912. 3 pp., 7 figs. *cdpA*.

Description and diagram of engine arrangement; one engine coupled to shaft, other connected to shaft engine through heaters and piping and transmitting its power to the shaft engine. Also description of producer.

DIESEL ENGINE, THE LANGEN AND WOLF. *The Engineer* (London), July 26, 1912. 1½ pp., 2 figs. *dmpC*.

Engine made in sizes up to 600 h.p. for auxiliaries on shipboard.

EXPLOSIONS-GASTURBINE, WIRKUNGSGRAD DER, Hans Holzwarth. *Zeitschrift des Vereines deutscher Ingenieure*. June 15-22, 1912. 6¾ pp., 6 tables, 5 curves. *em*.

Efficiency of the explosion gas turbine. Reply of the author of Die Gasturbine to Professor Stodola's criticism of that book.

GENERATOREN BAUES, NEUES AUS DEM GEBIET DES. *Der praktische Maschinen Konstrukteur*, April 11, 1912, 3 pp., 5 figs. *cpA*.

Data about a producer gas generator with turning grates built by Schneider & Co., Le Creusot, France. Power required to turn grates, 6 h.p.; power required for blower, 4.5 h.p.; air pressure, 40 m/m.

GENERATOR GASFEUERUNG FÜR BRENNÖFEN DER FEINKERAMISCHEN INDUSTRIE, DIE. Walter Stampe. *Braunkohle*, May 24, 1912. 3 figs., 1 curve. *cepB*.

Discussion over the advantage of using producer gas fired ovens for baking fine ceramic articles.

INTERNAL COMBUSTION ROAD LOCOMOTIVES AND THE DAVIS PARAFFIN CARBURETTER, THE ROYAL AGRICULTURAL SHOW. *The Engineer* (London), July 12, 1912. 2 pp., 5 figs. *cdpB*.

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[1] Opinions expressed are those of the reviewer, not of the Society. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *h* historical; *m* mathematical; *p* practical. A rating is occasionally given by the reviewer, as *A*, *B*, *C*. The first installment was given in The Journal for May 1910.

MOTOR SHIPS AND SAFETY AT SEA. What may be expected in the near Future Judging from the Results of the Motor Ship *Selandia's* Maiden Voyage. *The Engineer* (London), July 19, 1912. ½ p. *cdpA*.

OIL ENGINES. *Power*, July 9, 1912. 2 1/3 pp., 2 tables. *ce*.

Data on operating costs, oil consumption, etc. From report of the Prime Movers committee of the National Electric Light Association.

PRODUCER PLANT OF THE FORD MOTOR CAR COMPANY, GAS, O. J. Abell. *The Iron Age*, June 13, 912. 6 pp., 2 figs. *d*.

SELANDIA'S MAIDEN VOYAGE, THE. *The Engineer* (London), July 19, 1912. ½ p. *dp*.

The motor ship made her first voyage of 21,840 miles carrying 9300 tons of cargo on a consumption of 9 tons of oil per 24 hours.

VERGASER VON MOTORFAHRZEUGEN, STUDIE ÜBER DIE, A. Heller. *Zeitschrift des Vereines deutscher Ingenieure*, July 6, 1912. 9 pp., 7 figs., 1 table, 8 curves. *dmp*.

A study of automobile carburettors.

ZWEITAKTMOTORS, UNTERSUCHUNG DES ARBEITSPROZESSES EINES, H. Scheit. *Zeitschrift des Vereines deutscher Ingenieure*, June 1, 1912. 9 pp., 10 figs., 6 tables, 9 curves.

An investigation of the two-cycle engine.

EMPLOYMENT BULLETIN

The Society considers it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for the Bulletin must be in hand before the 12th of the month. The list of men available is made up of members of the Society, and these are on file in the Society office, together with names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

0188 Young engineer, who has specialized in design and manufacture of stokers. Salary $2000. Apply through the Society.

0189 Assistant engineer, permanent position. Must be experienced designer of electric traveling cranes. Accurate and quick at figures; capable of systematizing engineering data. Give references as to experience and personal habits. State technical training and salary desired. Apply through the Society.

0190 New York concern desires first class chief draftsman, capable of handling efficiently a force of about twenty men. Should possess thorough technical knowledge and have special experience in design of centrifugal pumping machinery and steam turbines.

0191 Superintendent wanted for established and growing two-cycle marine motor factory, located in large city in the Middle West. Must be good mechanic, capable of designing jigs, resourceful and a producer. Salary $2000. Apply through the Society.

0192 Head draftsman wanted by firm manufacturing pumping machinery, employing 40 men in engineering department. Must have had a successful experience as an executive, either as chief draftsman or second hand, in a large modern plant, preferably along similar lines of work. A man of sanguine temperament, energetic, enthusiastic, and with initiative, is desired, thoroughly acquainted with up-to-date drawing room methods and system; a good designer and standardizer. Apply through the Society.

0193 Man capable of perfecting not only actual working device but following through the shop and making a success of a folding machine for inserts in connection with newspapers, magazines, etc., capacity of 10,000 an hour. Salary $1500 and upward. Apply through the Society.

0194 Competent engineer to take charge of two boilers of about 30 h.p. and drive ice machines. Must be experienced in foundry work. Salary

$100 per month, with free quarters. Sign agreement for three years. M. Shaik Ipoh, Straits Settlements, British India.

MEN AVAILABLE

479 Mechanical engineer or superintendent for large corporation; experience along broad mechanical and steam power lines; executive, with ability to handle large number of men, or will consider proposition to act as consulting engineer.

480 Mechanical engineer, 40 years of age, American, has had both practical shop and technical training; is expert in the up-to-date manufacture of interchangeable parts, resourceful in design and process for reducing the cost and increasing production, good organizer and system man with excellent executive ability, expert in the design and manufacture of commercial vehicles, both gasolene or electric, Edison or lead storage batteries. Can furnish best of references.

481 Member, possessing broad general experience in shop, drafting room, and executive positions, desires connection with large manufacturing company where ability in devising labor-saving appliances, machinery, and methods of manufacture will be of value. Part of remuneration preferably based upon percentage of savings effected in reduction of manufacturing cost.

482 Junior, M.I.T. graduate, three years experience in building, design and construction. At present employed as engineer by large industrial concrn. Desires position with firm of constructing engineers or building contractors.

483 Mechanical engineer, 33 years old, married, graduate of leading technical school, wishes position as manager of small growing plant. Three years experience as machinist apprentice and two years as draftsman, four years in industrial and production engineering, installing cost, shop and production systems rearranging and equiping plants for increased efficiency, standardizing products, operations and tools. Now employed as general superintendent of factory employing 150 to 200 men.

484 Mechanical and electrical engineer, ten years experience in the design, construction, operation and management of steam and water power plants for industrial and public service companies, desires permanent position with consulting or contracting engineers in or near New York.

485 Associate member, age 38 years, Lehigh University graduate, now in charge of the construction of a 9000-kw. lighting plant, will be open for employment in an executive capacity, as manager, engineer or superintendent, upon the completion of the present work, October or earlier. Sixteen years varied experience along mechanical, electrical and civil engineering lines, involving design, supervision, construction, operation, purchasing, etc., in connection with electric railways, power and industrial plants, etc. Good executive. Salary $3000.

486 Superintendent of power department or power plant. Wide experience covering twelve years in construction and operation. Expert on

modern steam turbine plants. Practical, systematic and economical. With present concern five years.

487 Member, with twelve years experience in jobbing and contract work, covering foundry, machine, boiler and plate, and structural steel. Experience from draftsman to superintendent. At present, sales engineer for large power house engineers and contractors.

488 Position as works engineer wanted by member, 34 years of age, of broad experience in power plant design, construction, operation, heating systems, building construction and general factory engineering.

489 Position desired by executive engineer, member, 36 years old, seventeen years broad experience in designing, building, erecting and operating, power generating, hydraulic, electric, crushing, grinding, conveying, power transmission and machine tool machinery; concrete and civil engineering. Past seven years executive positions in industrial plants, competent organizer and handler of labor, able to secure from both labor and machinery maximum economic output. Average salary for past four years, $5000. Present location, South West, excellent references from past connections.

490 Technical graduate, 30 years of age, Junior member, at present employed in locomotive shops, five years designing and estimating costs in the manufacture of railroad and contractors equipment. Four years with engineers and contractors on power and industrial plant design. Will consider position as assistant engineer salesman or branch manager. Personal interview can be arranged in New York or Philadelphia. Minimum salary at beginning, $1800.

491 Associate, aged 35, at present employed as checker in drawing room, desires position of some responsibility in Philadelphia or suburbs. Sixteen years experience in civil, structural and mechanical lines, mainly the last. Employed as draftsman, squad foreman, chief draftsman and checker at rolling steel and pipe mills, by-product coke ovens and on chemical apparatus, steel cars and locomotives.

492 Position as superintendent, assistant superintendent or mechanical engineer of manufacturing plant, twenty years experience covering coal-handling machinery, agricultural implements and house-heating apparatus. At present employed. Details and references furnished to those interested.

493 Mechanical engineer, technical graduate, desires position as assistant superintendent or designer. Can furnish good references.

494 Motor-truck engineer open for position as transportation expert and efficiency man; competent to investigate and report upon delivery system. Eight years' continuous experience in this industry in designing, constructing, testing motor trucks, and writing on truck subjects.

495 Position desired with mechanical trade paper by editor and writer of several years' experience. Sufficient previous practical training and technical education. Knowledge of cuts, printing, make-up and make-ready, all phases of publishing work. Competent to bear responsibility and direct others.

496 Technical graduate and member desires position as works manager or superintendent. Wide experience in manufacture of high-pressure boilers. At present employed.

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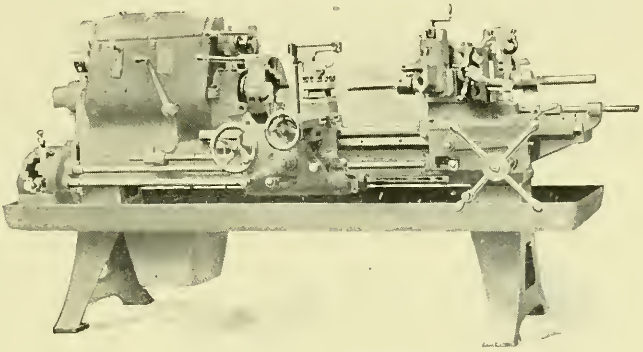

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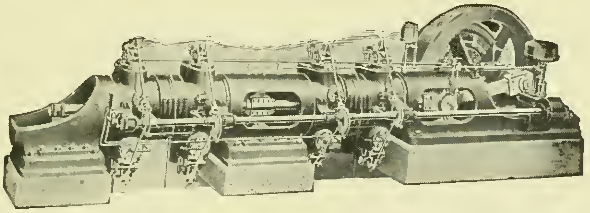

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Multi-stop and Double Turners

FIG. 1 illustrates the advantage of the *double stop* for each position of the turret, and the *double* adjustment of each turner. This piece has six finished diameters and six shoulders, and is turned by only three turners, which occupy only three positions on the turret. This not only leaves the remaining positions free for other tools, but it saves the operator the time and energy required to run the turret slide back each time.

All this is obtained without complication, and without introducing any features that are annoying when not in use.

In addition to the double stop for each of the six positions of the turret, we have an extra stop, consisting of a pin which may be dropped into any one of the six holes at the rear of the turret slide. This makes it possible to borrow five extra stops for any one of the tools, and gives to this tool seven length or shoulder stops, and leaves one stop for each of the remaining tools.

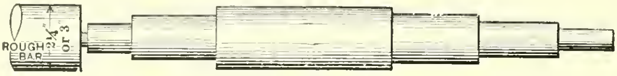

FIG. 1

The illustrations, Figs. 2 and 3, give examples of what one tool can do in this machine on chuck work, when we take advantage of the seven length stops and the seven shoulder stops of the cross-feed head.

Of course, in general practice three or four stops for one tool are all that will be needed, but since the modern cutting steels have greater durability, there is nothing lost by giving each tool all the work it can do.

Outer face and all shoulders and diameters accurately finished to independent stops by one tool. When roughing and finishing cuts are required, the roughing tool can be set near enough to use the same stops that are accurately set for the finishing tool. When an extra tool is used to give a roughing cut it is set as indicated by dotted lines in Figs. 2 and 3.

We find it difficult to illustrate all of the classes of work that can be turned out by this machine, but a little thought will suggest

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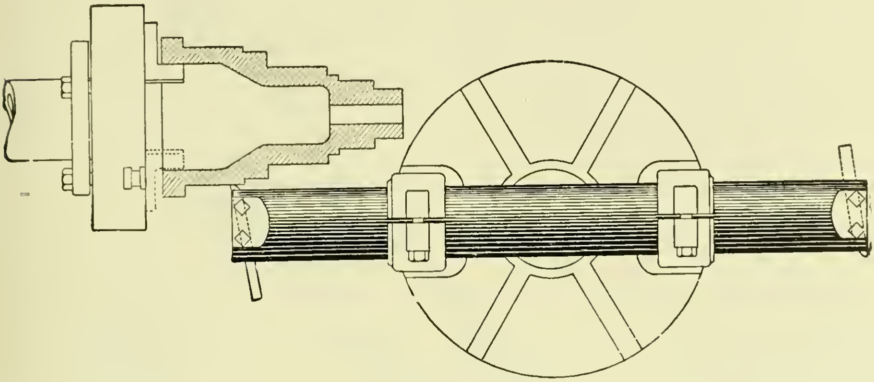

Fig 2

many forms that may be readily handled in bar and chucking work, both steel and iron, on account of the many provisions for bringing both turret and cross slide up to fixed stops; either by power feed or by hand.

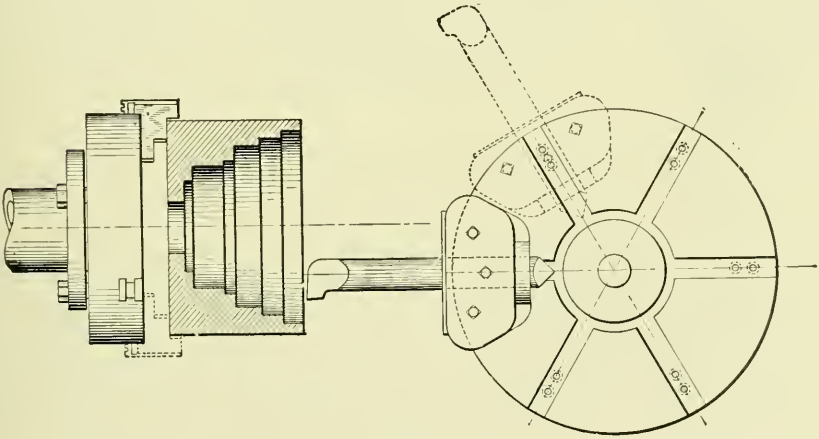

Fig. 3

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The table below shows the results of observations in representative foundries in different parts of the country. They are not "test" figures, but results accomplished under every-day working conditions.

Size of Cope	Time in Peining and Ramming By hand	By Sand Rammer	Ratio of Reduction	Per cent. time saved
12′x18″x4″	5 min.	1 min.	1:5	80
12′x18″x10″	10 min.	1½ min.	1:6.6	85
6′x3′x6″	20 min.	3 min.	1:6.6	85
6′x6′x8″	35 min.	8 min.	1:4.4	77
8′x6″x6″	1 hour	10 min.	1:6	83
7′x3′x12″	1 hr. 30 min.	16 min.	1:5.6	82
15′x30″x16″	2 hours	27 min.	1:4.4	77
12′x7′x16″	2 hr. 12 min.	34 min.	1:3.9	74
87″x159″x10″	4 hours	40 min.	1:6	83
19′x99″x15″	8 hours	1 hr. 30 min.	1:5.3	81

Note that these figures cover the entire job—both ramming and peining. **The saving of 74 to 85 per cent above indicated is fairly representative of what "Crown" Rammers will do for you, when they are intelligently used in average foundry work.**

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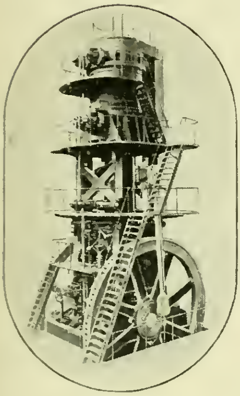
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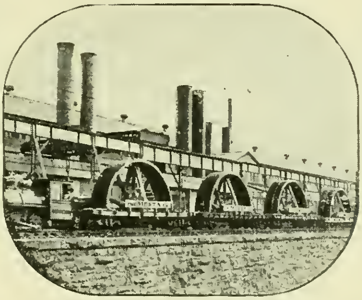
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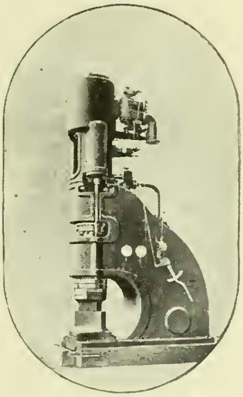
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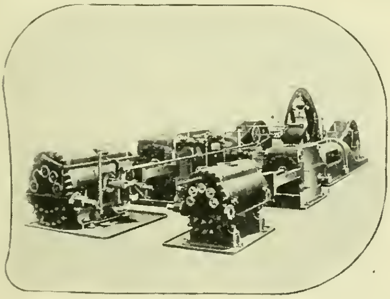
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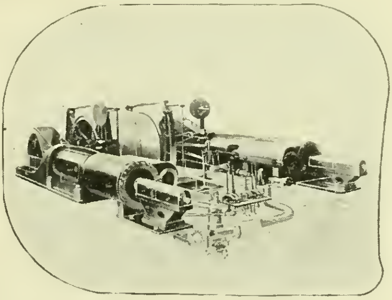
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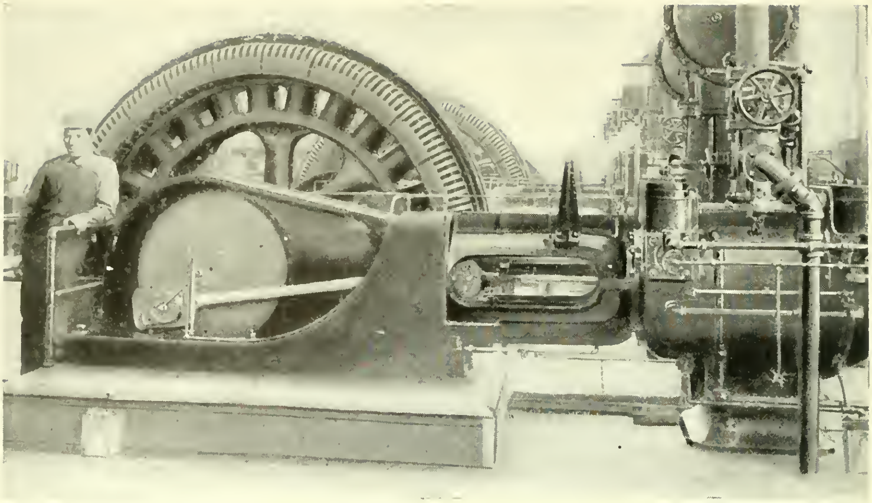

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THE accompanying illustration shows the coal pocket of Boyce and Barnes now owned by the Curtis Blaisdell Company, and was originally of the inclined-boom tub rig, which has since been changed to a Mead-Morrison automatic grab on mast and gaff hoisting outfit. Steam is the motive power of the hoist, and by means of an automatic or weight balance railway, the different grades and variety of coal are distributed to their proper bins. The grab bucket has a capacity of one ton and easily handles from 500 to 700 tons per day.

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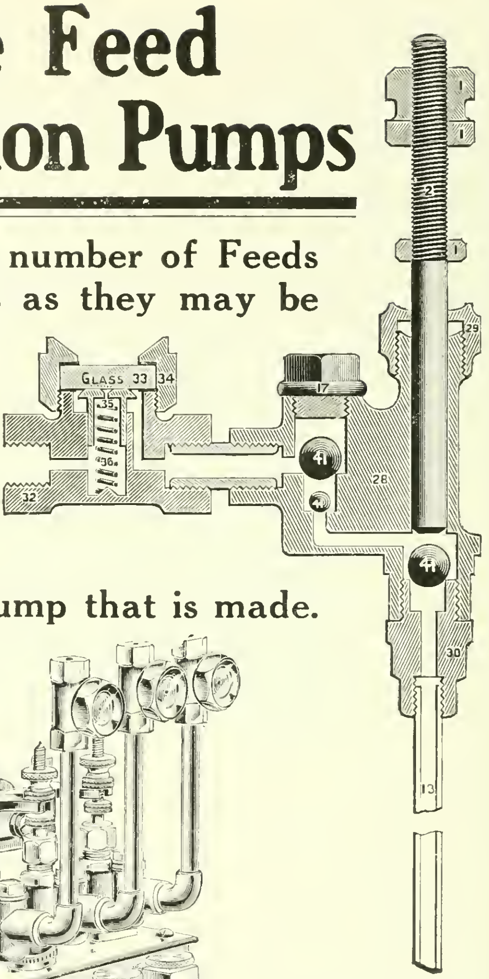

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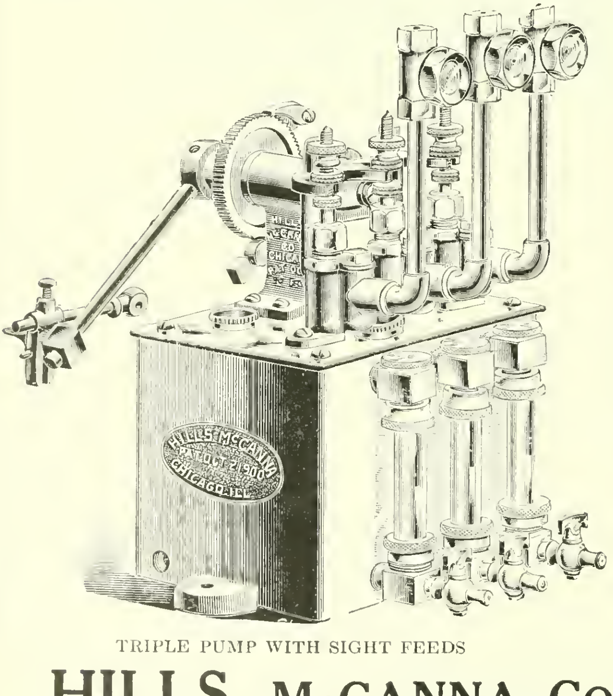

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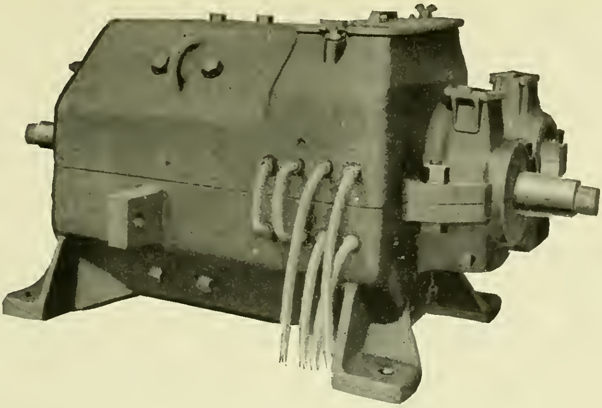

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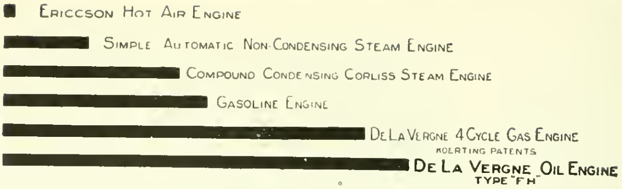

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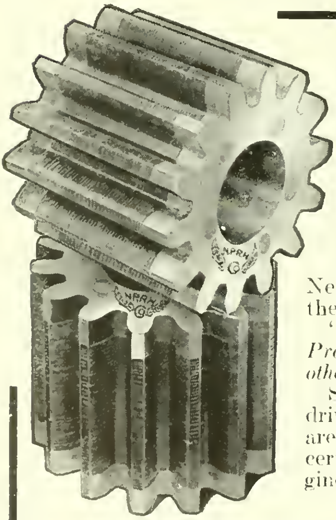

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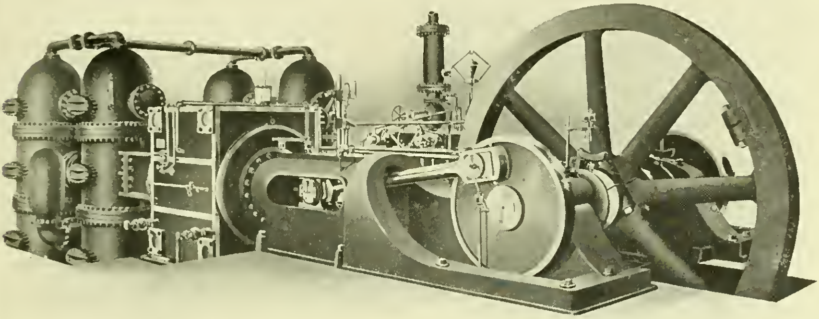

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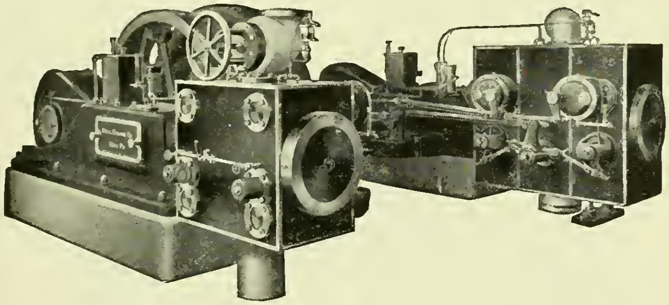

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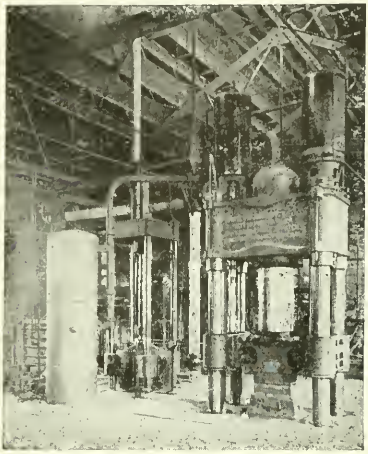

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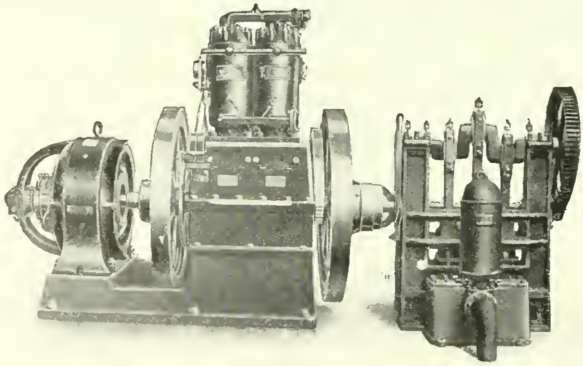

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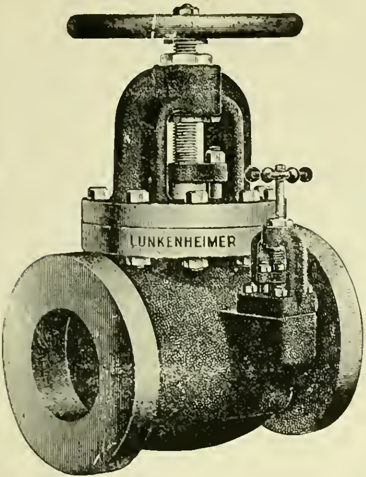

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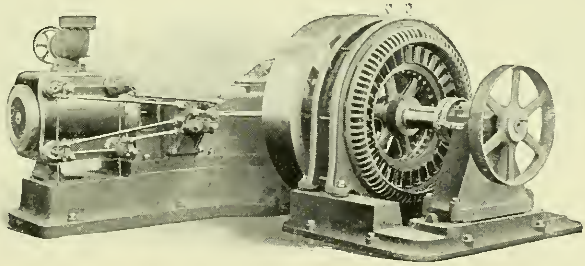

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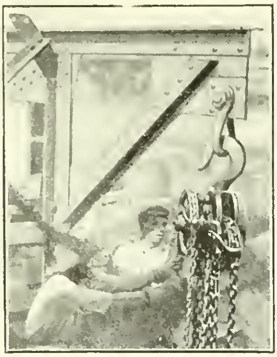

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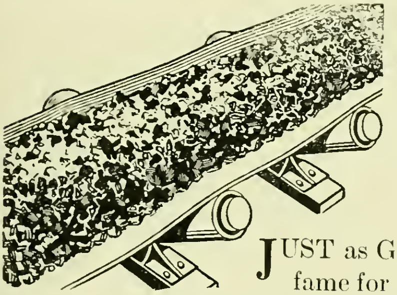

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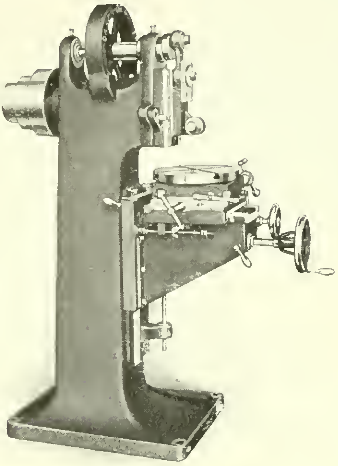

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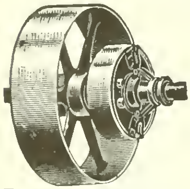

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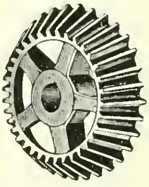

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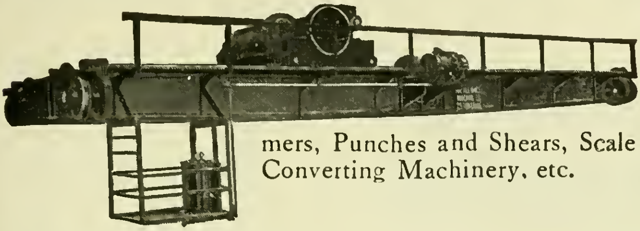

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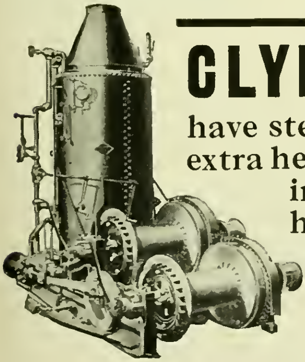

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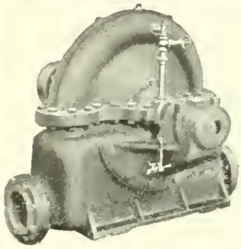

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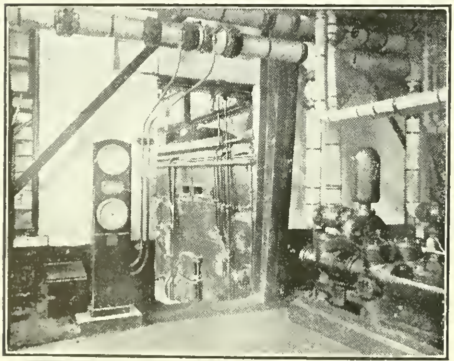

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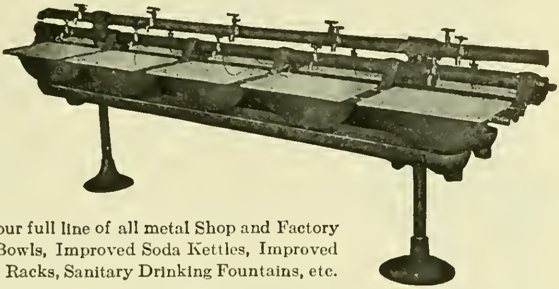

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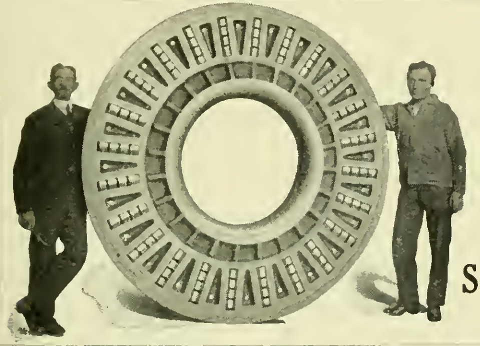

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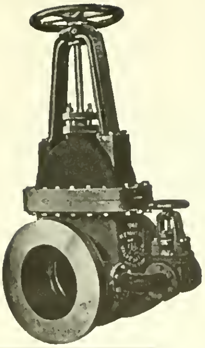

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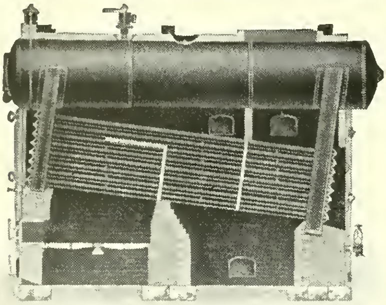

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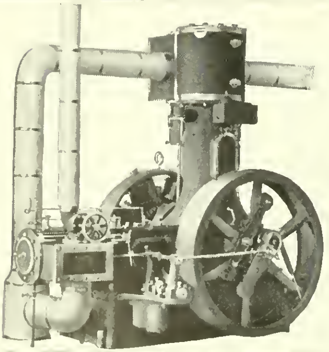

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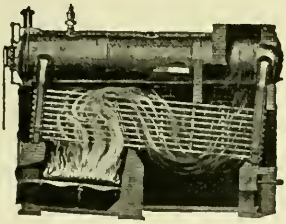

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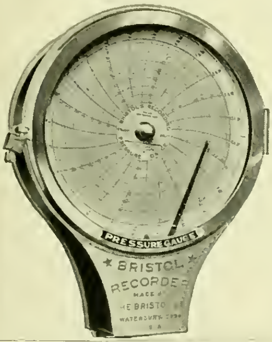

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Rockwood Paper Frictions have proven their unquestioned superiority. You will find our booklets regarding Transmission of Power by Belts and Friction Transmission desirable additions to your engineering library.

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Modern and Approved Appliances for the transmission of Power. Shafting, Couplings, Collars, Hangers, Pulleys, Belt Tighteners, Friction Clutches, Rope Driving Equipments.

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ROCKFORD DRILLING MACHINE CO.

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MANUFACTURERS OF

UPRIGHT DRILLS, GANG DRILLS, AND LATHES.

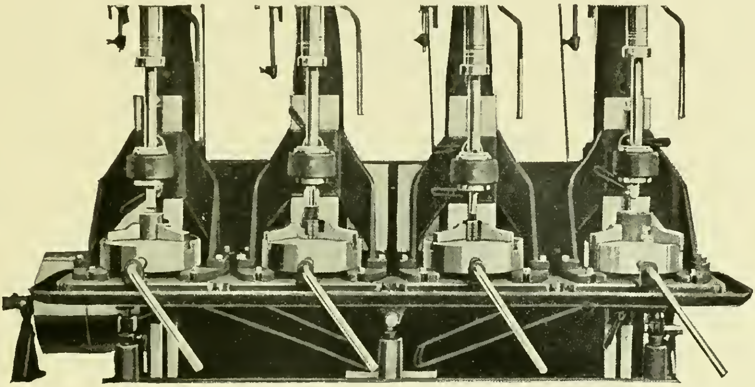


Illustration shows method of jigging and tooling a four spindle 23-inch gang drill for boring, facing one end, and reaming bushings.

ROCKFORD GANG DRILLS

The vertical drill press may be used to great advantage for many operations ordinarily performed by the horizontal method, at the same time involving less investment and insuring equal or greater production and simple chucking and handling.

The secret lies in the proper jigging and tooling of the machine, gauging the number of spindles per operator to the time required for the longest operation and the loading and unloading of pieces.

Every second spent by the operator hanging onto a lever or waiting for the power feed to run through the cut is a second wasted, and sufficient spindles should be used per operator to keep his time fully occupied.

The reason for ganging spindles in groups is to place as many in front of the operator as he can keep busy, thus utilizing time that otherwise is wasted.

We manufacture a complete line of upright drill presses from 12″ to 37″ swing, and build upright drill presses in gangs of 2, 3, 4 and 6 spindles on the 14″ and 20″ sizes, and 2, 3 and 4 spindles on the sizes larger than the 20″.

We construct four types of gang drills.

1st. Where a single table is used and the distance from spindle to table is the same in all cases.

2nd. Where a single table is used and the spindle bearings adjust to various heights. These two are of the box column type.

3rd. Independent columns having independent adjustable tables with spindle bearings stationary.

4th. Independent columns with adjustable tables and also adjustable spindle bearings.

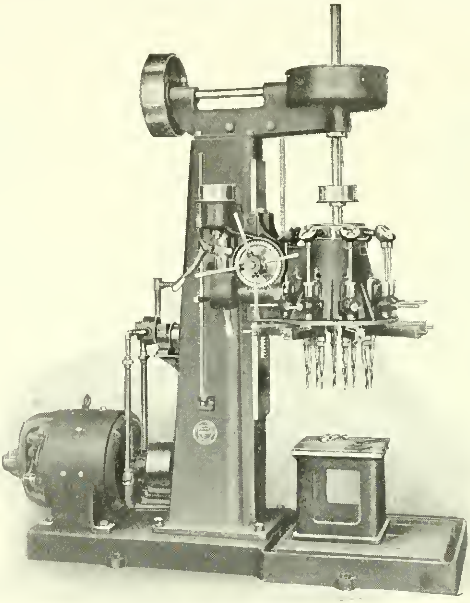
A booklet illustrating various methods of jigging and tooling Rockford Gang Drills will be sent free on request, and if you will send us a blue print of your work giving material, operations, and required production, we will submit estimates free of charge.

BAUSH MACHINE TOOL COMPANY

SPRINGFIELD, MASS.

New York Office: 50 Church St.

RADIAL AND MULTIPLE SPINDLE DRILLS



All Geared Multi-Spindle Drill

The *All Geared* Multi-Spindle Drill here illustrated was brought out to cover a range of work where universal joints were not equal to the heavy strains necessary to get full service from the best high speed twist drills.

The substitution of gears for the universal joints permits a design of sufficient strength to operate high-speed twist drills of any make to their maximum capacity.

The Spindles can be set to drill any layout desired whether circular, square, rectangular, or in a straight line, and may be adapted to meet special requirements when desired.

The speeds on belt driven machines will be from a three step cone and when motor driven a 2 to 1 variable speed motor will be used.

The feed mechanism on head is of the most approved type, having steel and bronze gears throughout; the head can be operated by hand or power and has automatic knock-off with quick return by hand.

Drill spindles have three changes of gear feed per revolution of spindle.

All gears are made of steel or bronze, hardened where necessary and properly guarded.

All bearings are bronze bushed.

Spindles are furnished with ball thrust bearings.

Oil pump and circulating pipes, with automatic relief.

Each spindle has individual oiling arrangement with flexible tube.

Head is counterbalanced by weight inside of column.

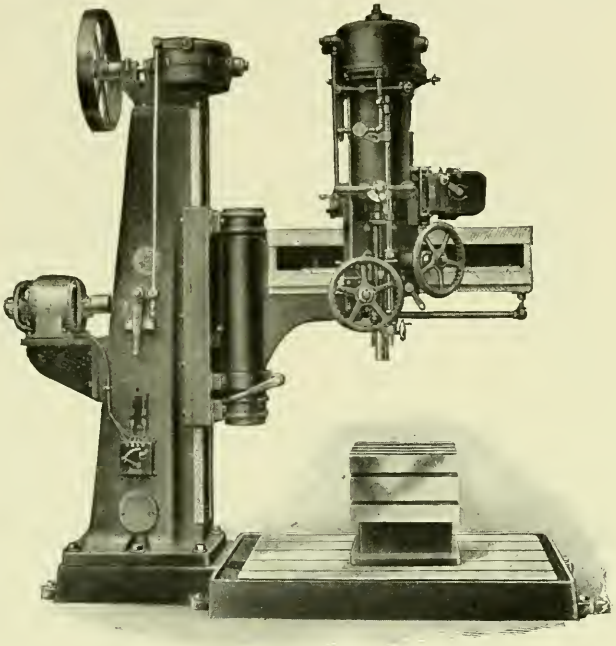
SPECIFICATIONS OF STANDARD MODELS

Model	No. of Spindles	Maximum Drill Used	Minimum Drill Used	Max. Circle with 8 Spindles	Min. Circle with 8 Spindles	Max. Sq. with 4 Spindles	Min. Sq. with 4 Spindles	Min. Dist. Center to Center of Spindles	Max. Dist. from Spindles to Bed	Width of Belt Double	Max. Height	Floor Space	Net Width
1-A	8	½″	¼″	12″	4½″	8¼″	2″	1¾″	42″	4″	8′-10¾″	3′x7½″	3600
4-A	8	1″	½″	18″	7¾″	12¾″	3¼″	3″	51″	8″	11′-9″	10′-9″x5′-2″	10900

BAUSH MACHINE TOOL COMPANY

SPRINGFIELD, MASS.
New York Office, 50 Church Street

RADIAL AND MULTIPLE SPINDLE DRILLS



Hydro-Pneumatic Radial Drill

The principle involved in the drilling head was worked out by the American Locomotive Company in anticipation of the now quite universal requirements for holes in boiler work drilled from the solid, instead of punching and reaming. The advantages are:

Direct application of a variable speed motor to rotate the spindle, in this way delivering nearly the whole torque of the motor at the point of the drill.

Feed by pneumatic pressure (at 80 lbs. per sq. in.) operating against an oil chamber, the flow from which determines the maximum advance of the drill. This combination gives a steady feed to the spindle and eliminates all danger of breakage from striking an extremely hard spot, or breaking through at the finish of the hole.

Other important features are, that the arm is equipped with a clamping device operated from the saddle, in fact all operating adjustments are on the saddle. All important bearings are either roller or ball bearing, and the spindle is high grade nickel chrome steel.

SPECIFICATIONS OF STANDARD MODELS

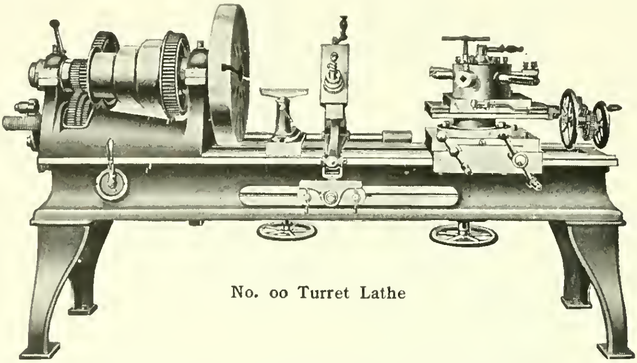
Model	Well Drill in Center of	Traverse of Head on Arm	Maximum Ht. from Nose of Spindle to Base	Minimum Ht. from Nose of Spindle to Base	Spindle Speeds	Feed, In. per Min.	Maximum Ht of Machine	Maximum Lgth of Machine	Maximum Width of Machine	Size of Working Surface	Net Weight including Motor and Table
4 ft.	111 in.	36 in.	62 in.	12 in.	50–	0 to 10	10′-9″	9′-7″	42 in.	36½x56½	7000
6 ft.	161 in.	56 in.	78 in.	23 in.	26–350	0 to 15	12′-6″	13′-8″	48 in.	42x83	12000

AMERICAN TOOL AND MACHINE CO.

ESTABLISHED 1845

BOSTON, MASSACHUSETTS, U. S. A.

LATHES, VALVE-MILLING MACHINES, OIL SEPARATORS



No. oo Turret Lathe

Turret Lathes (Nos. 00 and 0): Intended for heavy brass and iron work, and rod work up to 2-in. diameter. The forward motion of the turret can be accelerated by simply throwing the lever, and so is available instantly for either quick or slow operation.

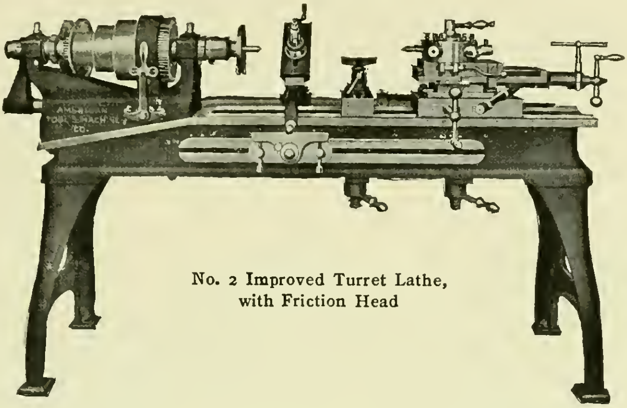
Cabinet Turret Lathes (Nos. 1 and 2): These lathes are exceedingly compact and firm, and not liable to spring and chatter. Especially suited for general service on "Chucked work." Head stock swivelled for boring taper holes with power feed.

Friction Head: For work requiring widely different speeds in its several operations, we have designed a friction head, so called. This combination enables the operator to use slow or back gear speeds by changing a lever.

Catalog containing full description, illustrations and specifications of lathes, sent on request.

No.	Actual Swing.	Length of Bed.	Spindle.	Takes Between Centers.	Fitted with		
					Back Gears	Screw Apparatus	Taper Attachm't
TURRET LATHES.							
00	26⅜	8 ft.	Hollow	46 in.			
0	24	8 ft.	Solid	46 in.	"	"	"
1	18	6 ft.	Solid	33 in.	"	"	"
CABINET TURRET LATHES.							
1	20½	7 ft.	Solid	34 in.	"	"	"
1 With Friction Head	20½	7 ft.	Solid	34 in.	"	"	"
2	18½	6 ft.	Solid	27 in.	"	"	"
IMPROVED TURRET LATHES.							
2	18½	6 ft.	Hollow	34 in.	"	"	"
2 With Friction Head	18½	6 ft.	Hollow	34 in.	"	"	"
IMPROVED LATHE							
1	17	6 ft.	Hollow	27 in.	"	"	"
SQUARE ARBOR LATHE.							
1	15	6 ft.	Solid	26 in.	"	"	......
2	13	5 ft.	Solid	27 in.	......	"	......
SET OVER AND BACK MOTION LATHES.							
2	13	5 ft.	Solid	27 in.	......	......	......
3	12	5 ft.	Solid	26 in.	......	......	......

AMERICAN TOOL AND MACHINE CO.



No. 2 Improved Turret Lathe, with Friction Head

Improved Turret Lathe: The headstock can almost instantly be raised to a level with that of the tail stock. The hollow spindle has a thrust bearing that runs continuously in oil, thus relieving the spindle of friction and wear.

Headstock fitted with self-oiling, hard metal, bronze boxes with hardened steel thrust and take-up, washers and adjustable caps.

Tailstock fitted with swivelling arrangement and quick motion device to top slide.

Screw apparatus, Fox design with six point star follower, hobb spindle revolves at half the speed of live spindle, insuring a strong and durable leading thread.

OIL SEPARATORS

Belt Driven

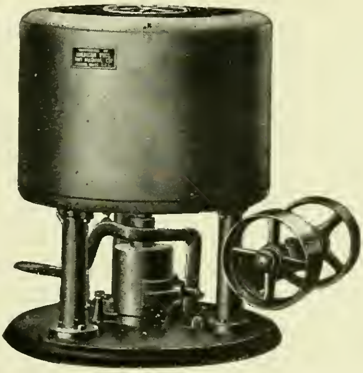
Made in two sizes. No. 1 has a capacity of 520 cubic inches, and will separate in from five to eight minutes, the time being regulated by the condition and quality of the oil used.

These same conditions apply to the No. 2 or larger machine, which has a pan capacity of 2,540 cubic inches, and is proportionately heavier and stronger. The time required to separate with the No. 2 is practically the same as with the No. 1, provided both are running at the same peripheral speed. The No. 2 separator (see cut), having about five times the capacity of the No. 1, is better adapted to bulky, light-weight chips or turnings, but is equally effective on ordinary work.

Electrically Driven Separator

The No. 1 separator is mounted and geared for electric driving. Electric power is now in such general use for machine-shop work that this type of separator meets a demand from shops so equipped.

Direct Current Motors furnished for any voltage.



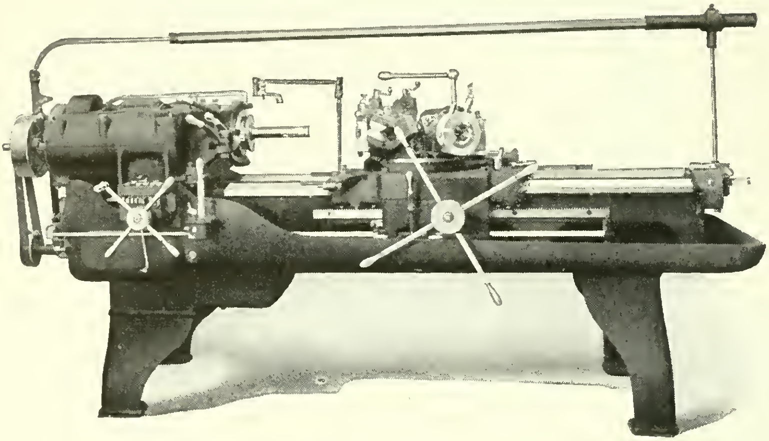
No. 2 Oil Separator

JONES & LAMSON MACHINE COMPANY

SPRINGFIELD, VERMONT, U. S. A.
77 Queen Victoria St., London, England

COST REDUCING MACHINES FOR LATHE WORK

THE HARTNESS FLAT TURRET LATHE



Single Spindle Type

The Hartness Flat Turret Lathe which was originally designed for bar work exclusively has been growing as a machine for Chucking Work. This drift toward chuck work commenced with the introduction of the Cross Sliding Headstock in 1904 and has been particularly well warranted, for by this feature its working range was extended to that of Chuck Work without departing from the original cardinal principle of stability. This was accomplished by the unusual departure of mounting the headstock on a cross slide, thereby gaining a double motion for each tool in the turret, without giving up the ideal feature of Single Slide plan.

The Single Slide principle provides the most absolute control of the work and tool. It has certainly not been equalled by any of the other schemes which have had to resort to the use of two or more slides between the tool and the bed casting.

On this plan has been carried forward the success of the Flat Turret Lathe which began back in 1890 in our works, and to the success of which we humbly acknowledge our present attainment.

All through these years we have aimed to build a machine on ideal lines, both mechanically and economically—and for ourselves and our customers. Our claim has been set forth in our book, Machine Building for Profit, and our machines have been described in the Hartness Manual of the Flat Turret Lathe. Although these books were in the nature of advertising mediums, the facts were stated with the best degree of impartiality that could be held by earnest advocates.

Our Flat Turret Lathe today stands without a rival as the machine that gives the best independent control of work and tool, (and you know that that is the most

JONES & LAMSON MACHINE COMPANY

SPRINGFIELD, VERMONT, U. S. A.
77 Queen Victoria St., London, England

COST REDUCING MACHINES FOR LATHE WORK

THE HARTNESS FLAT TURRET LATHE

Single Spindle Type

essential feature of a lathe for chuck work; if it were not, then you could do all your chuck work on a drill press at a great saving in cost of installation and labor cost per piece).

It is the only machine that combines high efficiency with adaptability. (The adaptability is the essential feature in any plant in which there is a progressive change in the design of its product. It is the feature that facilitates getting the work done when it is wanted. It is the feature that keeps down the cost of special tools, the immense capital tied up in parts, and the loss entailed by that large stock when on account of progress in design these parts have become obsolete.)

It is the simplest machine. (Simplicity cannot be overrated. The problems of the engineering world are intricate beyond the reach of the human mind, and any machine that is needlessly complex is a hindrance to progress. A simple machine for accomplishing an important work is one of the greatest blessings of the present day. It eliminates the worry about how to do the work. It facilitates the comprehension of the process. It brings the machine within the reach of the workman's understanding without his having to study a lifetime to know how to use it. In fact the simple machine is the machine of greatest value provided it meets the economic conditions.)

It likewise possesses many other unique characteristics which were most easily acquired in the development of this type of lathe. Easy, because we were the first in the field, and we took the natural means and selfishly patented them. This has not led to a public loss. It has led to the production of the greatest number of these machines by one manufacturer and this has made it possible to give a machine of the highest value at the lowest price. Machine builders know what gains can be made by this process, yet even they are frequently chagrined at our success.

But all these facts are well known to our friends who have bought our machines, and many of our friends who have bought others. **Just now, however, we are announcing something that may not be generally known.**

WE HAVE MADE TWO NEW DEPARTURES

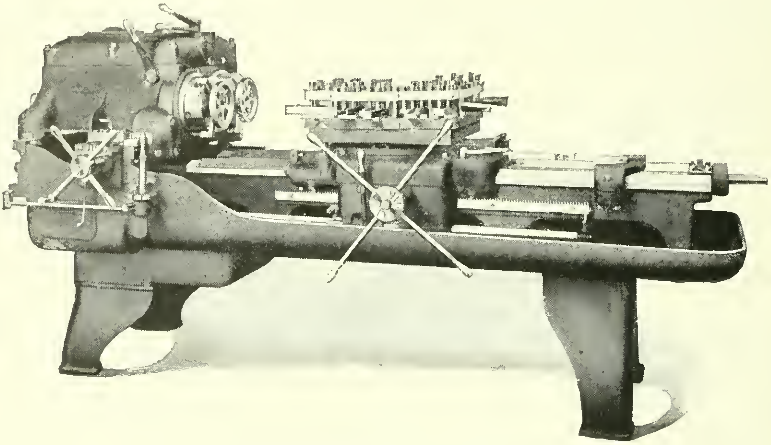
First, we are making the Flat Turret Lathe in two styles. The Regular style you know well. (It is made in two sizes. The smaller one is called the 2¼ x 24 size. It handles bar work up to 2¼ inches diameter of the bar from which the work is turned, and chuck work up to 12 inches in diameter and from 1 inch to 8 inches or 12 inches long, according to the character of the piece. The larger one is known as the 3 x 36 Flat Turret Lathe, and it handles bars up to 3 inches diameter, and chuck work up to 14 inches and from 1 inch thick to 12 inches or 18 inches long, according to the character of the piece.)

JONES & LAMSON MACHINE COMPANY

SPRINGFIELD, VERMONT, U. S. A.
77 Queen Victoria St., London, England

COST REDUCING MACHINES FOR LATHE WORK

THE DOUBLE SPINDLE FLAT TURRET LATHE



Double Spindle Flat Turret Lathe

The New Machine will be known as The Double Spindle Flat Turret Lathe. It is designed primarily for chuck work. It may be used either as a single spindle or a double spindle lathe. When used Single it can carry a chuck of 17-inch diameter, and when both spindles are used the two chucks are 9-inch diameter. This double spindle is another step toward highest output per dollar of investment. It possesses many unique features especially contrived to get out large lots of work. For small lots of work the single spindle lathe should be used, or this machine should be used, with one instead of two spindles.

The Two Spindle Machine gets it output by the use of two sets of tools. It carries on two similar cuts at the same time. Since these cuts are exactly the same, the speeds and feeds may be the most advantageous, the long cuts being taken on both pieces at the same time, and the short cuts are also taken simultaneously. This, of course, has a distinct advantage over the machines of multi-spindles in which one set of tools must operate on all of the pieces, for in such machines the total time is a multiple of the longest operation divided by the number of spindles, whereas in the present scheme the total time is only the sum of all the operations divided by two, the short one counting only for the exact time required instead of counting as much as the longest one.

The simplicity of operation and the similar character of the two cuts results in a greater saving than is apparent at first glance. It will be readily appreciated by those who have been exasperated by the excessive compromises that must be made in some instances in producing work on the multi-spindle turret machines in which only one set of tools is used. The loss of time on some kinds of work is a serious handicap, especially on work which requires a great difference in time in the performing of the several operations.

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77 Queen Victoria St., London, England

COST REDUCING MACHINES FOR LATHE WORK

THE DOUBLE SPINDLE FLAT TURRET LATHE

The advantages of the Double Spindle Flat Turret Lathe will be found to be many on many kinds of work, but of course it is designed more for manufacturing conditions of large repetition work than for conditions which call for quick change and small lots.

It gains its advantage over all other means not only on account of its double cuts of a kind, but **it combines with the double spindle the advantages that go with a hand-operated lathe,**—such as highest cutting speeds and feeds. These, of course, cannot be attained by an automatic machine that is not constantly under the watchful eye of the operator. The difference in speeds makes an important difference in output.

The Double Spindle Lathe also requires a little more care in setting the second set of tools so that they may act in harmony with the first set, but even this is minimized by the auxiliary adjustment provided for many of the tool holders; and another restriction is that with this doubling of tooling there is necessarily some restriction of the range of work, but it is nothing compared with the natural restrictions of other machines that are offered for manufacturing conditions. **Hence this machine and our Regular Single Spindle Lathe still maintain the leading position as adaptable broad range machines,** and these machines possess this desirable characteristic not by giving up their stability or convenience of control, but solely by well considered schemes of design, such as can only be brought forth by specialists.

For instance, let us look at the convenience of setting up the tools for turning a new piece of work on the Double Spindle. We will imagine we have a piece of chuck work about 8 inches in diameter, and, say about 4 inches wide, like a pulley, gear blank, or any similar piece which would require boring an inside hole, turning the outside diameter, facing the flange and hub, and perhaps turning the outside of the hub and inside of the flange and facing the web.

What is the most natural way to group these tools? What form of holding turret gives these tools the most stable support to resist the cutting stress? It seems to us that the most natural way to place these tools near the work previous to clamping them is to lay them down at the edge of a table with their cutting point projecting beyond that edge, just far enough to reach into or over the work, and that the best support for these tools would be such a table mounted solidly on a slide that will bring it up closely to the work so that there is no unnecessary overhang.

With such a thought in mind, we have given the Double Spindle Turret Lathe a square topped flat turret. The edges of its four sides may be presented one after the other to the work. Each edge may carry two sets of tools, the front set for the front spindle and the back set for the back spindle. These edges carry the tools which must have the most rigid and accurate support. (The four corners may be used for carrying drills, reamers, dies, taps or boring tools when working on one spindle at a time.)

Has there ever been presented a more ideal tool support for a Turret Lathe, both for meeting the most natural and convenient way of setting the tools for a new piece of work, and for the firmest control of these tools under working stress?

The means for clamping these tools to the top of the turret are the simplest, and, of course, the tools are likewise the simplest. Then, after that, the single slide scheme of operating reduces the uncertainty. For there is one handle that moves the turret carriage to and from the chuck, and another handle that moves the work-carrying headstock back and forth to get a change of diameter, and when either or both of these motions are made for one tool, we know that the other tool is similarly affected.

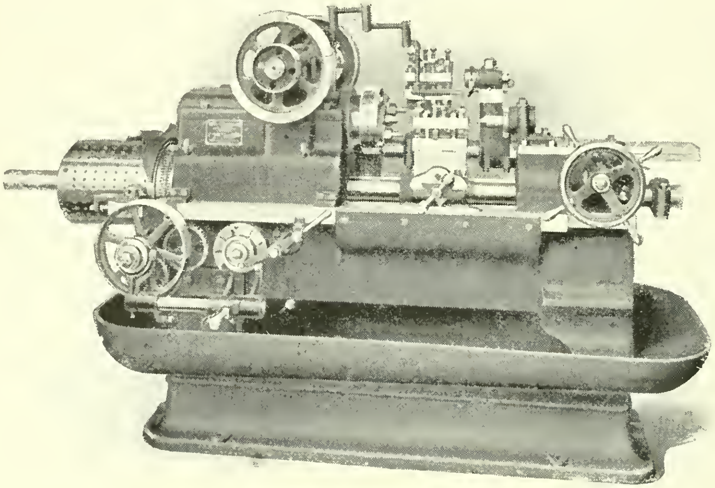
But we must go on with our story and give the second part of our announcement.

JONES & LAMSON MACHINE COMPANY

SPRINGFIELD, VERMONT, U. S. A.
77 Queen Victoria St., London, England

COST REDUCING MACHINES FOR LATHE WORK

THE FAY AUTOMATIC LATHE



The Fay Automatic Lathe

The Fay Automatic Lathe is now built and sold by the Jones & Lamson Machine Co. in conjunction with the Flat Turret Lathe.

The best industrialists know that it is not good practice to duplicate the number of handlings of a piece of work in the machine shop and we have won a great part of our good name by offering a way to do chucking work so that the whole piece came from the machine fully turned. That is, in many cases; in fact, in most cases where the piece could be slightly remodeled to meet our conditions we have been able in the Flat Turret Lathe to bore, turn, face, and in fact, perform a variety of intricate lathe operations on a piece of work without ever taking it out of the chuck. In turning a gear blank, for example, we have turned the outside and both faces of the rim, we have faced the hubs and bored the hole perfectly true, all with one setting simply by holding the gear blank by chuck jaws that have gripped outwardly on the inner edge of the rim between the arms of the blank.

So many ways have been found to fully turn a piece of work at a single setting, or at least to turn all of those surfaces that should be absolutely true with each other, that a great saving in cost of such work has been effected by our machine. And it is a saving that every Superintendent of a machine shop appreciates, for he knows that it is almost impossible to re-chuck a piece of work so that it will run true with the cuts first taken. So difficult has this been that many sacrifices in design have been made just to provide a way to do all of the turning operations on a piece at one setting, thus ensuring absolute truth of the work.

JONES & LAMSON MACHINE COMPANY

SPRINGFIELD, VERMONT, U. S. A.
77 Queen Victoria St., London, England

COST REDUCING MACHINES FOR LATHE WORK

THE FAY AUTOMATIC LATHE

But—there are pieces of work that must be turned on center point supports. That is, there are pieces of work that can best be supported on regular engine lathe center points. Some of these have no hole through them. In fact, some seem to combine the characteristics of both bar and chuck work and are so formed that there seems only one way to support them and that is on center points such as have been used in the good old engine lathe from the first days of its existence.

And it is to provide for this work that we have taken over the Fay Automatic Lathe. **It is offered for work that must be turned on centers as an auxiliary for the Flat Turret Lathe.**

The good feature about the center point support is that it is the means by which a piece may be taken out and a new piece put into a lathe in the shortest possible time. In other words, it is the quickest process of re-gripping a piece of chucking work. There are other quick ways, but they are lacking in accuracy. There is no "one way" for all pieces of work. There is, however, a one and only way that some pieces should be held, and it is for such pieces that should be held on centers that we are offering this machine.

The Fay Lathe is distinctly a center piece support automatic engine lathe for a certain range of work. It is in a class by itself, and is one of the simplest automatic machines for lathe work.

Although the stability of tool control is superior to many machines for automatic operation, it is not claimed that it approaches the ideal in this respect of the Flat Turret Lathe, but it is equally convenient for quick adjustment for any new piece of work, and it uses the very simplest of cutting tools. The success of this machine is attested by the batteries of them that have been installed by people who have had a chance to know of its merits.

It will be our purpose to furnish full information of this machine so that the work coming within its range may be done where it will be turned out at lowest cost and best advantage, for this is an important link in the lathe equipment of almost every manufacturing machine maker.

Remember, then, that the Jones & Lamson Machine Company of Springfield, Vermont, and London, England, are now building and selling the Double and Single Spindle Hartness Flat Turret Lathe, and The Fay Automatic Lathe for center point support work.

That the Hartness Flat Turret Lathes, Regular and the Double Spindle, are for chuck work as well as for bar work. That they cover chuck work up to 17-inch diameter, are built to give the best independent control of Work and Tools; the Highest Efficiency as profit producers from the investor's standpoint; the machines which give the greatest output when it is wanted, and which will produce almost any conceivable piece of work without special equipment.

The Regular Lathe is our standard machine for meeting the conditions found in machine building plants, such as automobile work, etc.

The Double Spindle meets the demand for the greatest output at expense of a little delay in "setting up" the machine for a new piece, (this delay being negligible on work which requires no change in tooling for periods of several days or more), and

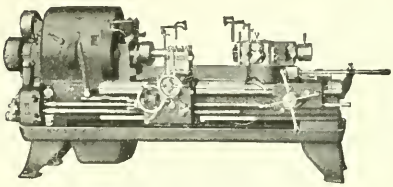
The Fay Automatic Lathe is for work that must be supported on centers.

THE WARNER & SWASEY COMPANY

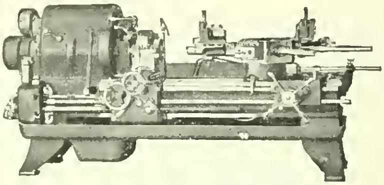
Works and Main Office: CLEVELAND, OHIO

Branch Offices: New York, Boston, Detroit and Chicago

TURRET LATHES TURRET SCREW MACHINES
BRASS-WORKING MACHINE TOOLS



No. 3A—Tools for Bar Work

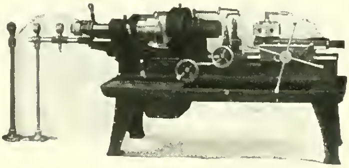


No. 3A—Tools for Chucking Work

HOLLOW-HEXAGON TURRET LATHES

For the rapid, accurate and economical production of lathe work; from bar stock, forgings and castings. Complete tool equipments for bar and chucking work.

Four sizes: Automatic Chuck capacity, 1½ to 4⅛″; length turned, 18 to 36″; swing over bed, 14 to 24″.

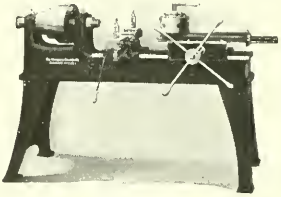


No. 6—Bar Capacity 2⅛″; Swing, 18″

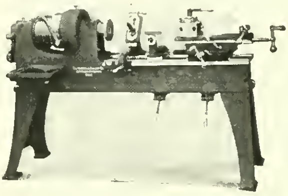
TURRET SCREW MACHINES

Five sizes—⅝ to 3⅝″ automatic clutch capacity; 10 to 20″ swing.

With or without automatic chuck; bar feed; automatic feed for turret; automatic feed for cut-off, etc.—every modern facility for rapid production.



Plain Turret Lathe



Universal Turret Lathe

TURRET LATHES AND BRASS-WORKING MACHINE TOOLS

Turret Lathes—14 to 24″ swing; Plain, Set-over or Universal turret; with or without Geared-friction Head, Automatic Chuck; Cut-off; Forming attachment; Chasing attachment, etc.

Automatic Boring and Tapping Machines; Valve Milling Machines; Key Lathes; Cock Grinders, etc.—for the manufacture of valves, cocks, fittings and similar work.

Equipments planned and estimates of outputs furnished upon request.

WM. SHELTON NICHOLLS

251 Broadway
NEW YORK, N. Y.

16 Church Street.
HOOSICK FALLS, N. Y.

TOOLS, MACHINERY, MACHINE SHOP EQUIPMENT

HERCULES PORTABLE CRANE HOIST

Patented Dec. 13, 1905

There are many firms whose work does not warrant installing an extensive system of overhead travelling cranes, but is nevertheless too heavy to handle by primitive methods.

There are other firms who already have overhead trolleys, travelling cranes, etc., which do not meet all the requirements of the plant. They are limited in scope to the length of the runways.

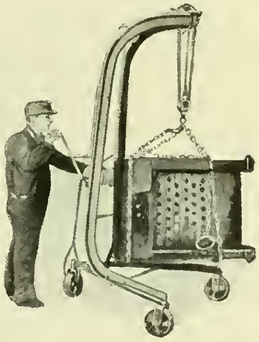
In both instances, there is a real need for our *Hercules Portable Crane and Hoist.*

This is a compact movable machine, with wheels on roller bearings. Operated by one man it will pick up a heavy casting, a shaper, or even a lathe, and move it to any point on the shop floor, or from one building to another.

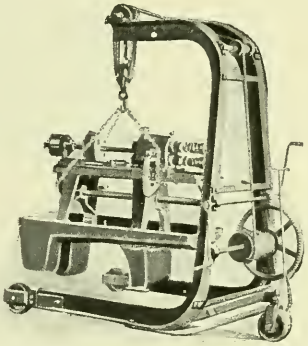
Its design is such that it can be used in many ways, and for countless purposes, and it is made in a variety of sizes to suit all requirements and pocketbooks. It is always ready, easily put aside when the work is done, and when not in use can be put in an out-of-the-way corner of the shop.

Of constant use in machine shop tending planers and lathes, etc.

The frame is of structural steel shapes, giving wonderful strength and reliability.



No. 1, Weight 285 lbs.
Capacity 1000 lbs.
Standard Pattern



No. 6-B. Wt. 1250 lbs. Capacity 6000 lbs. Low down Pattern

DIMENSIONS, WEIGHTS, AND CAPACITIES OF STANDARD SIZES

Size	Weight, Lbs.	Capacity, Lbs.	Height over all	Width over all	Length over all	Over-hang	Hoist	Price
1	285	1000	8 ft.	4 ft. 2 in.	4 ft.	1 ft. 11 in.	6 ft. 9 in.	$45.00
2	325	2000	8 ft.	4 ft. 2 in.	4 ft.	1 ft. 11 in.	6 ft. 9 in.	57.50
3	425	3000	8 ft.	4 ft. 2 in.	4 ft.	1 ft. 11 in.	6 ft. 9 in.	70.00
4	850	4000	9 ft.	4 ft. 8 in.	4 ft. 11 in.	2 ft. 6 in.	7 ft. 2 in.	90.00
5	940	5000	9 ft.	4 ft. 11 in.	5 ft.	2 ft. 9 in.	7 ft. 2 in.	105.00
6	1050	6000	9 ft.	5 ft. 2 in.	5 ft. 4 in.	2 ft. 9 in.	6 ft. 8 in.	125.00
6-B*	1250	6000	8 ft.	4 ft. 9 in.	6 ft. 10 in.	3 ft. 4 in.	5 ft. 8 in.	150.00

* Low down pattern.

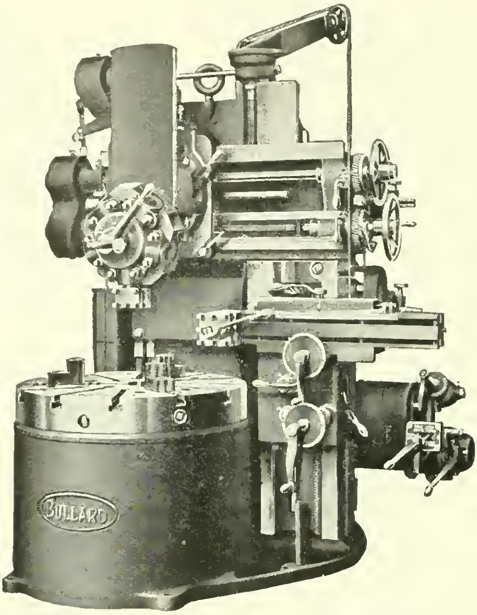
Write for Catalog

THE BULLARD MACHINE TOOL COMPANY

BRIDGEPORT, CONN., U. S. A.

THE VERTICAL TURRET LATHE

(New Era Type, Patented)



The Vertical Turret Lathe is a "boring, turning and facing machine," essentially original in design, in the development of which especial consideration was given to *the elimination of wasted time* in machine operation,—a decidedly large factor in productive capacity.

The basic feature of its design lies in the disposition, at a right angle to each other, of two universally and independently movable tool-carrying heads.

The Main Turret Head is mounted on a horizontal cross rail (as in the construction of the Vertical Boring and Turning Mill), and the Side Turret Head (similar to the tool carriage of a lathe) is supported by a *vertical* side rail; both rails forming a unit which is adjustably mounted on the Bed or Column, in the base of which is supported the table spindle and the entire driving and speed change mechanism—the whole forming a compact and self-contained unit of extreme strength and rigidity.

The unique disposition of the two Heads permits the *simultaneous machining of contiguous surfaces* without interference between Heads or Tools, and the *two Turrets* will hold in instant readiness all tools required for an entire sequence of operations.

The independent universal movements of *both* Heads simplifies the tool equipment required for a wide range and variety of work—the same tools (excepting reamers) being used for all work of the same type coming within the range of the machine.

OPERATIVE FEATURES

Which Eliminate Wasted Time

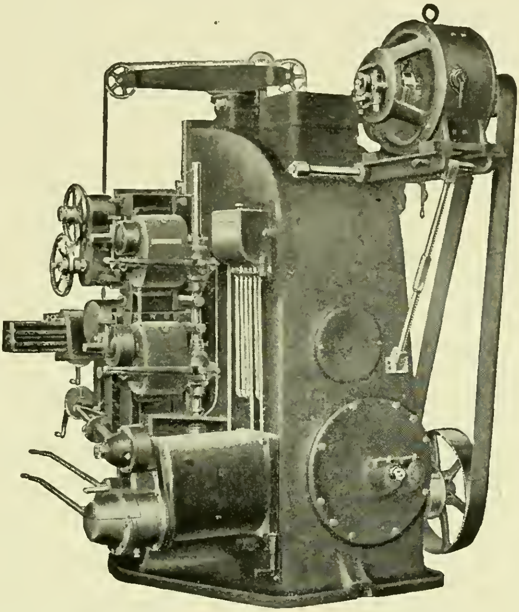
VERTICAL CONSTRUCTION—easy chucking with *gravity* as an aid.
QUICK SPEED CHANGE—the *right* speed easily *maintained*.

THE BULLARD MACHINE TOOL COMPANY

BRIDGEPORT, CONN., U. S. A.

THE VERTICAL TURRET LATHE

(New Era Type, Patented)



QUICK ACTING BRAKE eliminates *idle* table revolutions.

RAPID POWER TRAVERSE conserves the operator's energy.

CENTRALIZED CONTROL is *perfect* control—no long reaches or false moves.

LARGE MICROMETER DIALS are easily read in taking measurements.

OBSERVATION STOPS simplify the duplication of *various* sizes in repetition work.

STRUCTURAL FEATURES

Which Reduce Maintenance Cost

SINGLE PULLEY DRIVE at constant speed obtaining highest efficiency and longest life from belting.

MULTIPLE DISC CLUTCH running at constant speed maintains maximum efficiency regardless of table speed. Amply proportioned for heaviest duty and readily adjustable for wear.

ALLOY STEEL GEARS, throughout driving and feed train, heat-treated to 260,000 lbs. tensile and 245,000 lbs. elastic are practicably indestructible through either breakage or wear.

CONTINUOUS FLOW LUBRICATION of all bearings and gears insures highest power efficiency and immeasureably increased life of parts and obviates possibility of breakage and delay due to carelessness of operator in failing to fill any one of a "hundred and one" oil holes.

IMPROVED TURRET INDEXING, differing radically from standard construction, maintains accuracy of alignment and provides means of correction for *each* face should inaccuracy occur.

SIZES.

24 Inch and 36 Inch "Rapid Production Type."
36 Inch and 42 Inch "New Era Type."

THE KING MACHINE TOOL CO.

CINCINNATI, OHIO

30″ AND 34″ VERTICAL TURRET MACHINES.
VERTICAL BORING AND TURNING MACHINES 40″ TO 84″ INCLUSIVE

Having been designed after high-speed tool steels had fully demonstrated their possibilities, these machines are constructed to tax the best of them to their limit and have sufficient weight to not only withstand the strains imposed, but also to absorb the shock and vibration incident to machining under modern shop conditions.

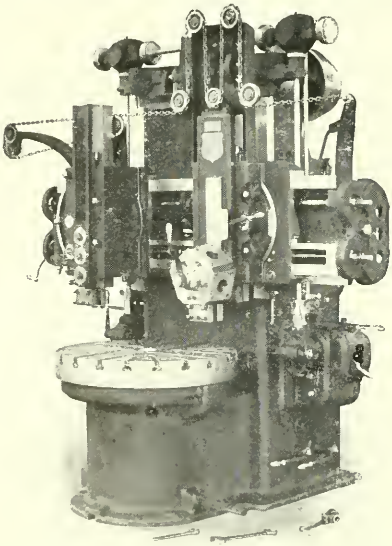
Every feature which has demonstrated its worth in modern boring-mill practice has been incorporated in the design of each machine, from smallest to largest, while no attachments have been added solely to serve as talking points and, consequently, these mills are free from needless complications and trappiness. The material entering into their construction is everywhere the best for the particular service demanded of that part, the workmanship is of the highest grade, all parts are made as nearly interchangeable as is practicable in machines of this size and excessive finish has been eliminated.

Care has been exercised to render all parts easy of access, easy to lubricate and easy to adjust; operating levers are so grouped as to require the minimum expenditure of effort on the part of the operator; gearing is fully protected by guards, the latter being hinged to swing aside at those points where access to gears is desirable. Motor-drive application was fully considered in making the original design and all sizes of machines are adapted to the use of any make of motor, either constant or variable speed—the latter having a maximum ratio of 2 to 1.

Original alignments are extremely accurate and provision is made for maintaining them. All sizes are entirely self-contained and should be placed upon solid, unyielding foundations in order to secure best results and most quiet operation.

IMPORTANT FEATURES.

Power rapid traverse to heads in all directions, with table either at rest or in motion



Instantaneous feed-changing mechanism, whereby any feed may be obtained at will.

Single-lever control, whereby the rapid traverse may be engaged **in any direction** by one lever, independently of positive feeds.

Resilient drive which eliminates chatter and gear marks.

Mechanical belt shifter, operated by a single lever.

Adapted to motor drive of any style or make.

Foot-brake to table of larger machines.

Safety device to guard against overloading of feed mechanism.

Friction rail-elevating mechanism, noiseless in operation and overcoming objectionable features of tumbler-gears at this point.

Automatic and adjustable knock-out to all feeds, positive in action and working within extremely close limits.

THE KING MACHINE TOOL CO.

GENERAL DESCRIPTION

Wearing surfaces and bearings are of such proportions as to insure durability and to withstand the most severe service to which machines of their size and capacity should be subjected. All plane surfaces are scraped to surface-plates and cylindrical bearings are ground. All driving shafts, feed shafts and cross-rail screws are made from high carbon, open hearth steel; screws are carefully chased in special machines and worms are milled. Spur- and bevel-gear pinions are of steel, spur gears are cut from the solid and bevel gears are correctly planed. All screws and nuts, except those not likely to require adjustment, are hardened.

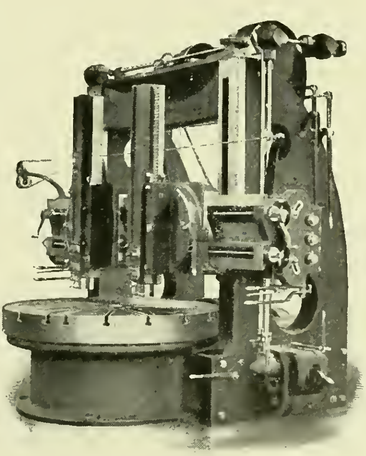
To eliminate the use of separate countershafts the upper cone pulley shaft is provided with tight-and-loose pulleys and is mounted on top of the machine; this arrangement gives all the advantages of the single-pulley drive, yet allows the use of efficient cone pulleys which, in connection with the mechanical belt-shifter, give rapidity of speed change equalled only by controller of a variable-speed motor. All machines are provided with four mechanical changes of table speed, two through friction clutches and two through positive clutches, and these, in conjunction with the changes obtainable through the cone pulleys, furnish from twelve to twenty table speeds, depending on the size of the machine.

Motor drives, employing either constant or variable speed motors, are easily arranged. If a constant speed motor is used it is belted to the upper cone-pulley or driving shaft, located on the top of machine, and the speed variations are obtained through the cone pulleys and the four mechanical speed changes of driving-head. If a variable-speed motor is employed, the cone pulleys and belt-shifter are dispensed with and the two to one speed-range of the former is obtained through the motor.

All machines are equipped with power rapid traverse to heads in all directions and also with what is termed a "rail-box," the office of which is to furnish change from vertical to horizontal movement, or vice versa, of the head which it controls by the manipulation of a lever and without the use of slip-gears at ends of rail. It also provides a safety point in the feed train and may be equipped with attachment for screw-and scroll-cutting.

The cross-rail is of the narrow-guide type and the heads are square-locked, machined from the solid without loose clamps and provided with taper gibs which are adjustable to compensate for wear. The tool-bars are provided with steel rack their entire length; they carry forged tool-posts and may he swiveled to either side of the vertical by worm-and-segment device.

The table is provided, on its under side, with a conical seat which rests upon a similar bearing on the spindle. This has a self centering tendency, reduces side-thrust on the spindle and is of such proportions that the maximum load borne is well within the limits prescribed for such service.

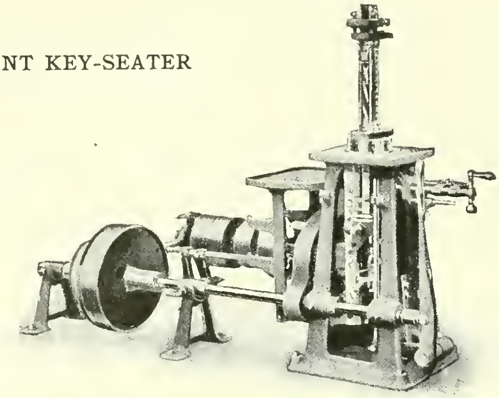


MITTS & MERRILL

SAGINAW, MICHIGAN

THE GIANT KEY-SEATER, KEY-SEAT MILLING MACHINES, A BORING MILL

THE GIANT KEY-SEATER



No. 2 Giant Key-Seater and Countershaft.

THE GIANT KEY-SEATER cuts perfectly true. Key-ways straight or tapered without regard to whether the hub is faced true or left rough as it comes from the foundry. Every job is quickly and accurately set and fastened by *its bore only.*

This means that in many cases it is possible to save the expense of facing a hub by using the Giant Key-Seater. The saving in money represented by this feature alone will soon pay for the machine.

A key-seat 6″ long, ½″ wide, and ¼″ deep can be cut in two minutes. This includes time of putting on and taking from machine.

The machines may be fitted to cut Key-seats in holes from ½ inch in diameter up to the largest size needed, and it is possible for the largest machine to operate in very small and long holes. In key-seating small work such as steel milling cutters ½ inch thick, eighteen or more pieces can be cut at one time. No other machine does this.

We can furnish cutters for special work, such as square holes, half-round key-seats, dovetail key-seats, double key-seats, or other special shapes such as are used in automobile work.

The Giant will finish two ordinary key-seats before one piece can be fastened, ready for key-seating on other styles of machines.

FLOOR SPACE, WEIGHT AND OTHER DATA

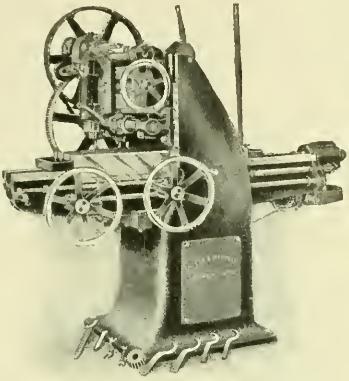
Key-seater	Stroke	Largest Post that may be used	Widest Cutter that may be used	Approx. Gross Weight	Floor Space
No. 0	7 inch	1½ inch	¾ inch	650 lbs.	2 x 3 ft.
No. 2	13 inch	2⅜ inch	1¼ inch	1500 lbs.	5 x 2 ft.
No. 3	16 inch	3⅜ inch	2 inch	1900 lbs.	5½ x 2 ft.
No. 3A	25 inch	3⅜ inch	2 inch	2000 lbs.	5½ x 2 ft.
No. 4	19 inch	3 11/16 inch	2½ inch	2100 lbs.	6 x 2 ft.
No. 5	25 inch	4⅞ inch	3 inch	4300 lbs.	6½ x 3 ft.
No. 6	31 inch	6 inch	4 inch	4800 lbs.	6½ x 3 ft.
No. 6A	38 inch	6 inch	4 inch	4800 lbs.	6½ x 3 ft.

MITTS & MERRILL

SAGINAW, MICH.

KEY-SEAT MILLING MACHINE

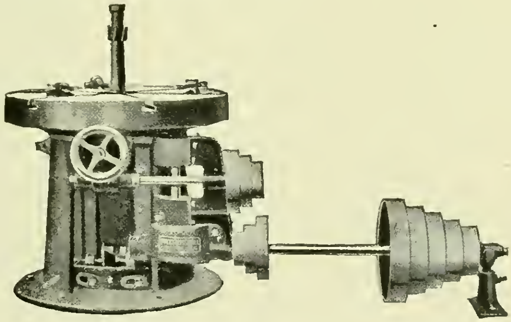
Our No. 3 Key-Seat Milling Machine is equipped with a vertical arbor for routing out the ends of key-seats. The machine is strongly geared and stiffly built for key-seating shafts up to 10″ in diameter, and for ordinary plain milling up to capacity of the machine. The hand wheels for the different feeds and movements of the machine are conveniently arranged and the weight of the crossrail and gearing is counterbalanced. The platen has basins on each end to catch the lubricant for which the rail serves as a reservoir. The vertical and cross feed have micrometers for fine adjustment. With the machine is furnished wrenches and an adjustable floor stand for shafting. Two smaller sizes are made as listed below. The No. 1 machine is only suitable for milling key-seats in shafting. Nos. 2 and 3 will do for plain milling also.



No. 3 Key-Seat Milling Machine
For 1 inch to 10 inch shafts.

Machine	Capacity	Table	Traverse	Floor Space Req.	Weight
No. 1	¾ in. to 3 in. dia. shaft	12 long	14	34 x 22	800 lbs.
No. 2	1 in. to 6 in. dia. shaft	26 x 12	34	36″ x 31 or 40	2800 lbs.
No. 3	1 in. to 10 in. dia. shaft	44 x 12½	43	74 x 36	4100 lbs.

BORING MILL



For Straight or Taper Holes up to 8 in. Diameter

We illustrate a Boring Mill which we manufacture and have used in our own shop for several years. It is designed for boring a class of duplicate or heavy work difficult to chuck in a lathe, and for the occasional job of large diameter. Driving shaft and pulley can be extended any distance required to swing the work on revolving table. Cutter bar is clamped in a saddle sliding on a vertical rail which is pivoted at the top and can be swung from the perpendicular to bore holes up to 2 inch taper per foot or more if required.

Capacity: will bore holes up to 8 inches diameter and 18 inches long.

Table: 35 inches diameter, 5 inches deep; has 6 T slots for ⅞ inch bolts. Height of table above floor, 33 inches. Bearing for table is a 7½° x 45° V bearing 3 inches wide, 18 inches diameter.

Vertical rail, 2 feet 9 inches long, 10 inches wide.

Saddle has bearing 14 inches long and a travel of 18 inches. Saddle holds boring bars up to $2\frac{15}{16}$ inches diameter.

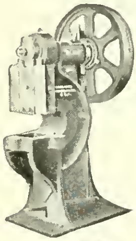
Floor space, 3 feet 6 inches by 6 feet 6 inches. Weight 2300 lbs.

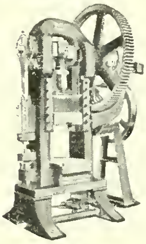
E. W. BLISS CO.

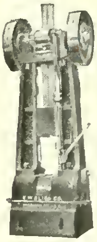
BROOKLYN, N. Y.

LARGEST BUILDERS OF

SHEET METAL WORKING MACHINERY







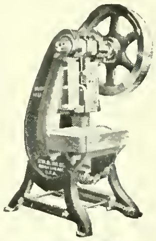
The most complete line of machines for sheet metal working in the world.

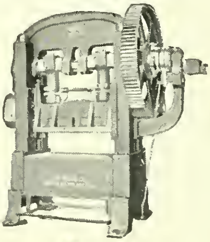
Presses for every ordinary kind of work and special machines for unusual requirements. Drop Forging machinery, Hinge and Butt machinery, Fork and Spoon machinery, Expanded Metal Lath machinery, Shovel machinery, Horse Shoe machinery, Minting machinery, Automobile Parts machinery, Spinning Lathes, Gang Slitters, Circle Shears, Perforating, Punching, Slitting, Shearing, Beading, Flanging, Crimping and Seaming machines.

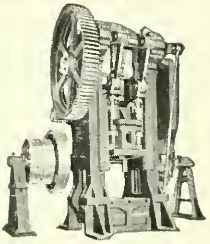
Complete equipments for the economical manufacture of Petroleum and Alcohol Cans, Fruit and Vegetable Cans, (Sanitary and Packers'), Meat Cans, Paint and Varnish Cans, Lard Pails and Butter Tins, and all kinds of Tin Cannisters, Boxes and Packages.

Machinery for manufacturing Soft Metal Tubes, Tinware, Enamelware, Aluminum, and Silverware, Metal Shingles, Metal Ceilings, Sheet Metal Furniture, Kitchen Utensils, Kitchen Boilers, Oil Stoves, Lamps, etc., etc.

We are also equipped for die work of every description.







Catalogues describing any of our lines will be sent on request

No. 1. "Bliss" Inclinable Power Presses.
No. 2. "Stiles" Power Punching Presses.
No. 3. "Bliss" Straight-Side Power Presses.
No. 4. "Bliss" Drop Hammers and Trimming Presses.
No. 5. "Bliss" Toggle Drawing Presses and Spinning Lathes.
No. 6. "Bliss" Foot and Screw Presses.
No. 8. "Bliss" Double Crank Presses.
No. 9. "Bliss" Machinery for Manufacturing Tin Cans (Spanish).
No. 10. "Bliss' Machinery for Manufacturing Pieced Tinware.
No. 11. "Bliss" Minting Machinery.
No. 12. "Bliss" Machinery for Manufacturing Electrical Parts.
No. 13. "Bliss" Drop Forging Machinery.
No. 14. "Bliss" High Speed Automatic Can Making Machinery.
No. 15. "Bliss" Railway Motor Gears and Pinions.
No. 16. "Bliss" Machinery for Manufacturing Automobile Parts.

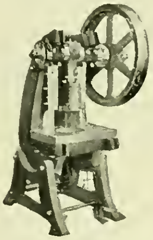
NIAGARA MACHINE & TOOL WORKS

BUFFALO, N. Y., U. S. A.

TOOLS AND MACHINES FOR SHEET METALS, PRESSES AND PUNCHES, POWER SHEARS, TINSMITHS' TOOLS, FORMING ROLLS, DIES, ETC.

Our line of tools and machines for working sheet metals is very extensive, the machines here illustrated comprising only a few representative types. We use nothing but the best of materials, and the castings are made in our own foundry. Superior facilities and a force of expert workmen enable us to produce work of the highest class.

Our products are presented on their merits and we solicit an examination of their efficiency and durability.

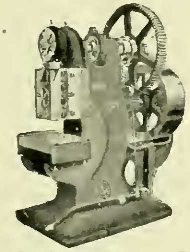


No. 6 Niagara Inclinable Power Press

INCLINABLE PRESS

These presses are unsurpassed in workmanship, convenience, and durability, and the range of work for which they are suited is almost unlimited, including blank cutting, punching, forming, and combination dies. The wearing surfaces are large and all parts well fitted.

9 sizes ranging from 500 to 7300 lbs. weight.



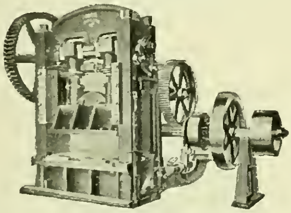
No. 38 Niagara Geared Power Punching Press

PUNCHING PRESS

These presses are adapted to heavy work, the design combining strength and compactness with convenience in handling the work. They are especially suitable for punching and cutting bars and heavy sheet metal; for operating cutting and forming dies required in the manufacture of hardware, cutlery, etc.

Five sizes geared.

Five sizes not geared.

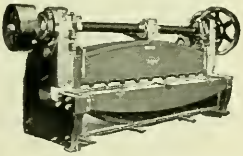


No. 615 Niagara Power Press Double Crank Type

DOUBLE CRANK PRESS

This press is intended for operating dies covering a large area and where great power is required. The speed is so regulated that in some cases the operators can remove and feed the sheets between two strokes without stopping the motion of the slide.

Will take work up to 13 x $56\frac{1}{2}$ inch area.



Niagara Power Squaring Shear No. 9126

POWER SQUARING SHEARS

This illustration shows one of our largest shears capable of shearing $\frac{1}{4}$-inch sheet steel, with a cutting length of 126 inches and a gap of 18 inches.

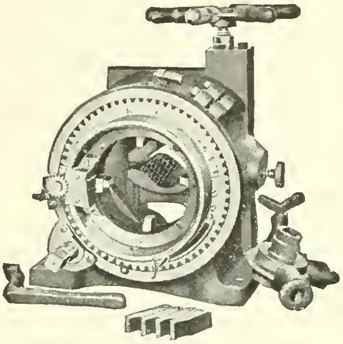
We make many smaller styles and sizes down to the No. 10 series intended for No. 22 iron and lighter.

Complete Catalog on Request.

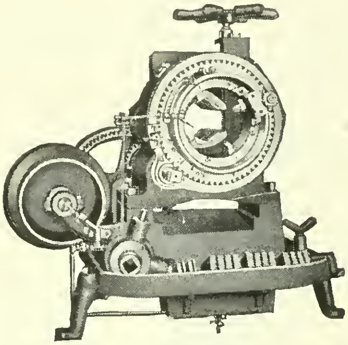
THE CURTIS & CURTIS COMPANY

93 Garden Street,
BRIDGEPORT, CONNECTICUT

PIPE CUTTING AND THREADING MACHINERY



Hand Machine



Power Machine

THE FORBES PATENT PIPE CUTTING AND THREADING MACHINE

In presenting this machine we would ask your attention to the following points:

It will save you its entire cost in a short time, in time and hard labor saved.

Our Pipe Machine is a complete tool, being all that is required to thread or cut off pipe.

It is much lighter than the old style of stock and the vise which must accompany it.

It is less expensive than the above combination.

It can be used in confined places.

It is particularly adapted to trench work.

Its dies can be sharpened by grinding without first drawing temper.

They draw back out of the way when the thread is cut, yet always cut standard sizes, and are adjustable to any variation of the fittings.

The shell can be adjusted to take up wear of gear.

Its dies are less expensive than solid dies, and last much longer.

Any one die of a set can be replaced when lost or worn out without replacing the entire set.

Its parts are interchangeable and can be duplicated when worn out, at a small cost.

MOTOR DRIVEN TYPE

This machine is equipped with a specially designed Die head which does away with all thumb screws for adjusting the dies, which are now clamped with one movement of a lever; it has self-centering vise.

BASE. Mounted on cabinet base with a motor for any current desired concealed within, and direct connected through triple compound cut spur gears on machine.

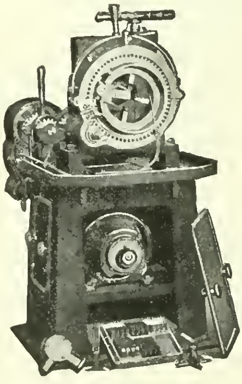
STRAINER. In center of the pan is located a strainer to separate the oil from the chips. It is thus drained, and can be used over again until unfit for use.

PUMP. Driven by gears. It is out of the way of the operator, and direct connected. No belts to annoy you.

GEARS. All are machine cut and entirely protected from chips or accidents.

MOTOR. Any current desired concealed in cabinet base, thus protected from oil or dirt, or breakage resulting from handling of long and heavy lengths of pipe and fittings.

Complete Catalog on Request



Motor Driven Machine

IDEAL OPENING DIE COMPANY

BEAVER FALLS, PA.

MANUFACTURERS OF SCREW CUTTING DIES

THE IDEAL OPENING DIE

Opens by power of cutting strain: The cutting strain in all opening dies has a tendency to revolve the head holding chasers and not the cam. It is obvious therefore that by allowing the head of our die to revolve, we convert the cutting strain into an opening power instead of depending on springs.

Head revolves instead of cam: The superiority of this construction over dies which depend on the life and variation of a spring for their successful operation, will be apparent to every mechanic. Spring actuated devices are notably unreliable, and apt to be out of order when most needed.

Faulty alignment overcome: We have designed in our die an equalizing driving clutch which compensates for any imperfection in alignment of either the turret or the material being threaded.

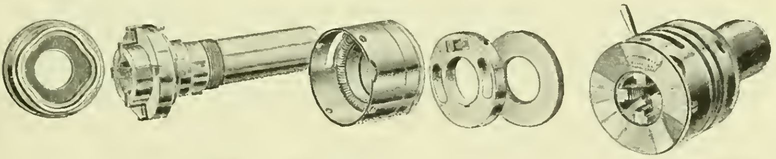
Chasers supported: The support for chasers in the Ideal Die puts an end to the tipping evil. They are supported by a hardened cam directly over and slightly preceding the point of strain. This chaser support together with the equalizing driving clutch, eliminates taper threads and bad work generally.

Four chasers instead of three: The advantage of four chasers over three is obvious. With four chasers you have the cutting points opposite, while with three chasers, as soon as one chaser becomes dull, it crowds the metal between the other two, thereby producing work that is not round.

Half as many parts: The Ideal Die has few parts and is easy to take apart and assemble.

Detachable Shank: The detachable shank makes it possible to use the same die in different sized machines.

Left and Right Hand Threads: The same die will cut either left or right hand threads; the former by simply changing index from right hand to left hand, and using left hand chasers.



Size	¼ inch	½ inch	¾ inch	1 inch	1¼ inch
Diameter of head	1 11/16 in.	2¼in.	2 15/16 in.	3 9/16 in.	4 11/16 in.
Length of head	1⅝ in.	1⅞ in.	2½ in.	2¾ in.	3½ in.
Diameter of shank	⅝ or ¾ in.	⅞ or 1⅛ in.	1¼ or 1½ in.	1½ or 1¾ in.	1¾ or 2¾ in.
Length of shank	2 in.	2½ in.	3 in.	4 in.	4¼ in.
Length over all	3⅝ in.	4⅜ in.	5½ in.	6¾ in.	7¾ in.
Capacity	3/32 to ¼ in.	⅛ to ½ in.	¼ to ¾ in.	⅜ to 1 in.	½ to 1¼ in.
Chasers in set	4	4	4	4	4
With 1 set of chasers	$25.00	$30.00	$40.00	$50 00	$65.00
Extra shanks, each	4.00	5.00	6.00	7.00	8.00
Standard chasers per set	1.25	1.50	2.00	2.50	3.00
Special chasers per set	2.00	2.50	3.00	3.50	4.00
H. S. Steel Extra	1.00	1.00	1.00	1.00	1.00

One set of Standard pitch chasers, either U. S. V. or Whitworth, right or left hand furnished with die.

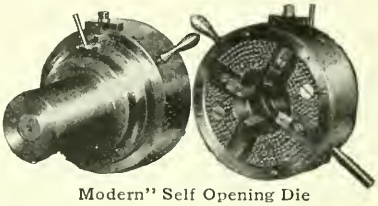
For sizes desired not listed in table above send specifications.

MODERN TOOL COMPANY

ERIE, PA., U. S. A.

SELF-OPENING DIE HEADS, ADJUSTABLE SOLID DIES, TAP AND DIE HOLDERS
CHASER GRINDERS, FRICTION AND POSITIVE DRIVE MAGIC CHUCKS
TAPPING ATTACHMENTS AND MODERN GRINDING MACHINES
MAKERS OF SPECIAL TOOLS AND FIXTURES

"MODERN" SELF OPENING DIES

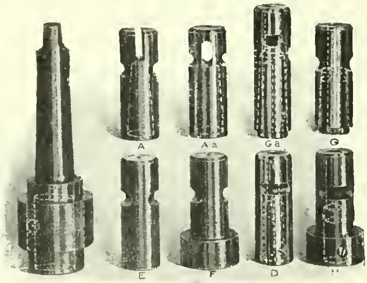

Modern" Self Opening Die

The advantages of the "Modern" Die are many. The Die opens automatically when the thread is cut. No reversing of the machine. The return is 2½ to 1. No danger of injuring either the thread or Die. 50% to 80% in time and wear to belts and countershaft is saved. The "Modern" is the one Die having a steel cam to hold Chasers in place directly over point of duty, making it impossible for Chasers to bell at the mouth and preventing irregular or taper thread. When a large amount of stock is to be removed or precision required, a roughing and finishing attachment will be furnished.

"Modern" Chasers are all cut on a hob and are made either of carbon or high speed steel.

"MAGIC" CHUCK EQUIPMENT

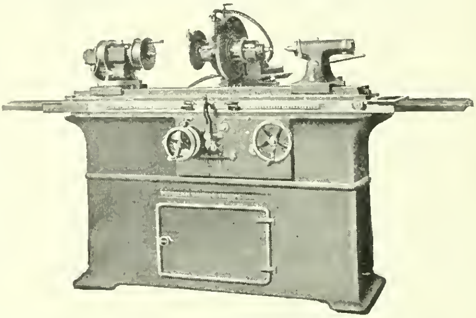
For the rapid changing of tools in drill press, lathe, screw machine, etc., without stopping the machine, practically converting a single spindle machine into a multiple spindle one, with as many tools as you may have operations. Try it and save labor cost.



Magic" Chuck and Collets

"MODERN" GRINDING MACHINES

Much unnecessary weight is eliminated and perfect rigidity and freedom from vibration secured by the scientific distribution of material in "Modern" Grinding Machines. Generous bearings for all moving parts reduces the possibility of wearing and prevents absolutely all chatter. Centralization of working parts secures perfect transmission of power to the working point. Accessibility of the working parts for care and attention by the operator makes minimum loss of use, cost of up-keep and depreciation. Improved operative features insure large production and precision work.


"Modern" Grinding Machine

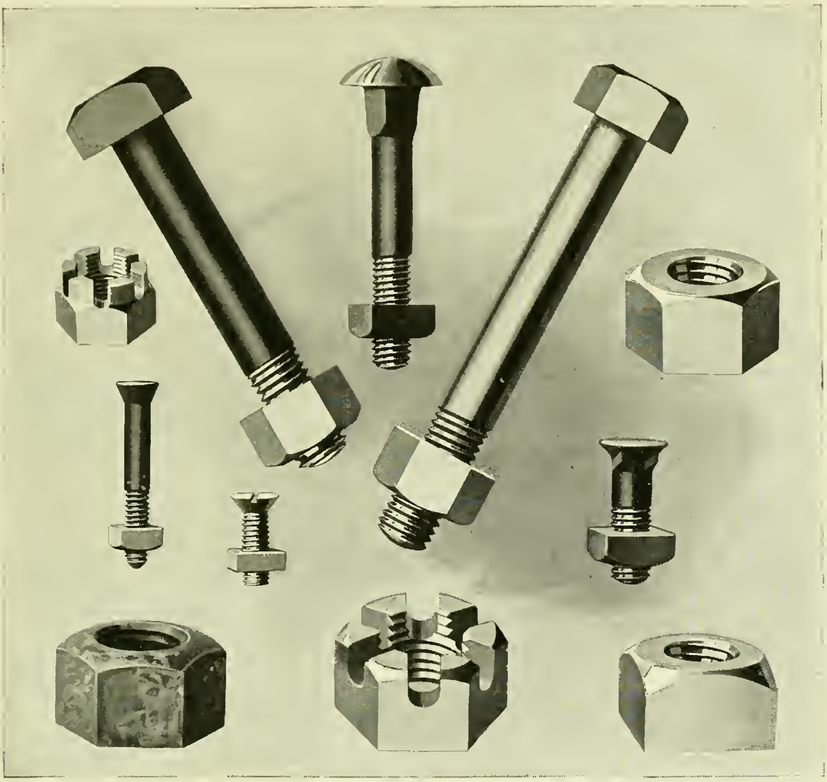
ALL "MODERN" TOOLS GUARANTEED TO BE SATISFACTORY TO THE USER

Complete information concerning products will be mailed interested parties upon request.

RUSSELL, BURDSALL AND WARD BOLT AND NUT COMPANY

PORT CHESTER, N. Y. ROCK FALLS, ILL.

BOLTS AND NUTS



Manufacturers of

All kinds of

- Carriage Bolts
- Machine Bolts
- Coupling Bolts
- Stud Bolts
- Tap Bolts
- Plow and Cultivator Bolts
- Stove Bolts
- Tire Bolts
- Rivets and Special Bolts of all descriptions
- Cold Punched, Chamfered and Trimmed Hexagon and Square Nuts
- A.L.A.M. Plain and Castellated Nuts
- Master Mechanics' Castle Nuts
- Semi-finished, Full Finished and Case Hardened Nuts

Our Trade Mark:

"EMPIRE"

signifies a certain standard of excellence that invites your investigation.

RHODE ISLAND TOOL COMPANY

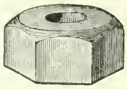
PROVIDENCE, RHODE ISLAND

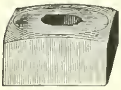
Established 1834

COLD PUNCHED, CHAMFERED, TRIMMED AND DRILLED SQUARE AND HEXAGON NUTS; SEMI-FINISHED, (TAPPED AND FACED) NUTS; FINISHED CASE HARDENED NUTS; CASTLE NUTS; MACHINE BOLTS; TAP BOLTS, ROUGH AND MILLED; COUPLING BOLTS; CAP AND SET SCREWS; LARGE HEAD CAP SCREWS; STUDS, ROUGH AND MILLED; CHAIN LINKS; WASHERS; PRESS WORK; FORGINGS; EYE BOLTS; TURN BUCKLES; WRENCHES; SCREW MACHINE PRODUCTS.

We are particularly fitted to furnish Screw Machine Work and Special Bolts and Nuts.

Chamfered, Trimmed, and Drilled Nuts are made by cold punching special material and have smoothly trimmed sides.



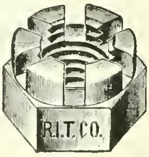


They are made Square and Hexagon and can be furnished tapped, if desired.

Semi-Finished Nuts are the same as the above, tapped and faced with washer finish on the bottom.

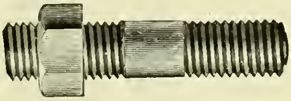
Finished and Case Hardened Nuts are finished all over with round top and then case hardened, giving a mottled effect.

Castle Nuts are semi-finished with three slots across the top for cotter pins.



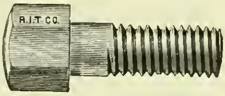
RHODE ISLAND TOOL COMPANY

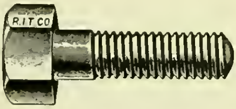
Machine Bolts are furnished with Square or Hexagon Heads and Square or Hexagon Nuts. Interchangeability is an important factor and this is guaranteed on all bolts.



Studs are made Rough or Milled. The material from which these are made is especially adapted for this work and was obtained after exhaustive tests. It is uniform in quality and its physical properties give it the strength and toughness which are so essential in this class of material.

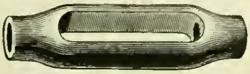
Set Screws and Cap Screws are made of special screw stock and particular attention is given to finish. Round or Cup Point Set Screws and Hexagon Head Cap Screws are carried in stock in large quantities.





Large Head Cap Screws are also carried in stock. These screws have a head which is United States Standard in size giving a greater bearing surface. This size of head also fits a Standard Wrench.

Machinists' Wrenches, Single, Double End and Set Screw are furnished with milled openings, semi-finished or finished and case hardened. The Semi-Finished Wrench is made with milled openings and case hardened with the heads polished. Special Wrenches can be made to advantage if blueprints are submitted.



Drop Forgings are made to samples or prints and prices will be quoted upon request.

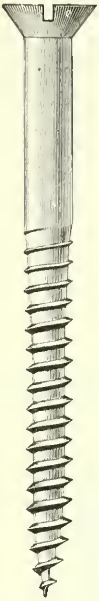
All of the above articles are fully set forth in catalogue showing lists and complete line of products and will be mailed to any one interested.

AMERICAN SCREW COMPANY

PROVIDENCE, R. I.

MAKERS OF WOOD SCREWS, MACHINE SCREWS, STOVE BOLTS, TIRE BOLTS, RIVETS, ETC.

Flat Head



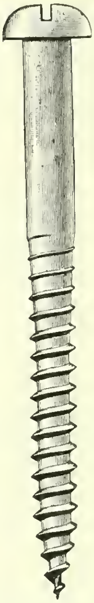
Oval Head

WOOD SCREWS

Flat and Round Head Wood Screws are regularly made in Iron in the following sizes, and in Brass in sizes of approximately the same variety; other kinds of Wood Screws are made in the sizes commonly used.

Drive Screw

Round Head



Length.....	1/4	3/8	1/2	5/8	3/4	7/8	1	1 1/4	1 1/2	1 3/4
Min. Dia...	0	0	1	1	2	2	3	3	3	5
Max. Dia...	4	9	12	14	16	16	20	24	24	24
Length.....	2	2 1/4	2 1/2	2 3/4	3	3 1/2	4	4 1/2	5	6
Min. Dia...	5	5	5	6	6	8	8	12	12	12
Max. Dia...	24	24	24	24	26	26	30	30	30	30

Intermediate diameters advance as follows: No. 0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 20 22 24 26 28 30

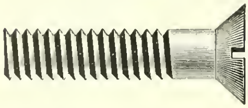
MACHINE SCREWS

Flat, Round, and Fillister Head Machine Screws are regularly made in Iron in the following sizes, and in Brass in sizes of approximately the same variety:

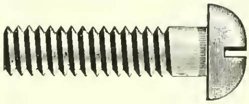
Length.....	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2
Min. Dia...	2	2	2	2	2	2	2	2	2	3	4	4	4	4	4
Max. Dia....	10	14	16	24	24	24	24	26	34	34	34	34	34	34	34
Length.....	1 3/4	2	2 1/4	2 1/2	2 3/4	3	3 1/4	3 1/2	3 3/4	4	4 1/4	4 1/2	5	6	
Min. Dia....	6	6	8	8	8	8	10	10	12	12	12	14	14	16	
Max. Dia....	34	34	34	34	30	30	30	30	30	30	30	30	30	30	

Intermediate diameters advance as follows: No. 2 3 4 5 6 7 8 9 10 12 14 16 18 20 22 24 26 28 30 34

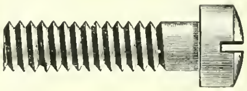
Flat Head



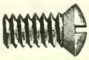
Round Head



Fillister Head



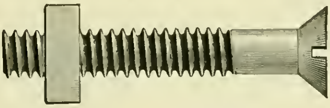
Diameter No.	2	3	4.5	6	7	8	9.10	12	14
Threads per in.	48.56 64	48.56	32.36.40	30.32.36	30.32	30.32.36	24.30.32	20.24	18.20.24
	16.18	20 22	24	26.28.30	34				
	16.18.20	16.18	14.16.18	14.16	13				

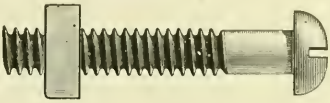
Regular Side Knob Screws  are 3/8 inch No. 9, 24 thread.

See also next page.

AMERICAN SCREW COMPANY

STOVE BOLTS


Flat Head


Round Head

Flat and Round Head Iron Stove Bolts are regularly made in the following sizes:

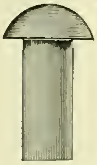
Diameter	1/8	5/32	3/16	7/32	1/4	5/16	3/8	1/2
Min. Length	3/8	3/8	3/8	1/2	1/2	3/4	3/4	1
Max. Length	2	2	6½	6½	6½	6½	6½	3

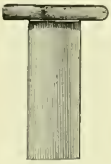
The length advances by eighths of an inch from 3/8 to ½, then by quarters to 6½.

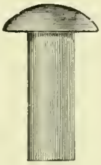
STOVE RODS

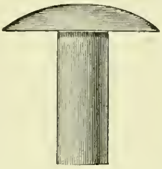
Stove Rods are the same as Stove Bolts in every respect excepting length. They are regularly made in Iron of 3/16 and ¼ diameter in length from 7 to 40," advancing by halves of an inch.

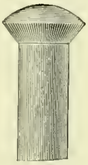
RIVETS

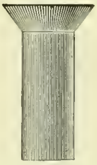

Round Head

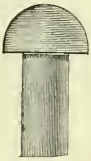

Flat Head


Truss Head


Wagon Box Head


Oval Counts'k.


Countersunk


Wheel Head

Cold-headed Rivets are made in great variety of styles and sizes up to 7/16 in diameter and 6 in length.

MEASUREMENTS

The length includes the head of Flat Head Screws, Stove Bolts, and Stove Rods; excludes the head of Round and Fillister Head Machine Screws and Round Head Stove Bolts and Stove Rods; includes the countersink of Oval Head Screws and about half the head of Round Head Wood Screws, but the practice with regard to Round Head Wood Screws is not uniform with all makers.

The length of Rivets is exclusive of the head for all styles with a right angle under the head, and inclusive of the countersink for countersunk heads.

The diameter of Screws is measured by the American Screw Gauge, the equivalent in inches being:

0	.0578	5	.1236	10	.1894	15	.2552	22	.3474
1	.0710	6	.1368	11	.2026	16	.2684	24	.3737
2	.0842	7	.1500	12	.2158	17	.2816	26	.4000
3	.0973	8	.1631	13	.2289	18	.2947	28	.4263
4	.1105	9	.1763	14	.2421	20	.3210	30	.4526
								34	.5053

The diameter of Rivets is measured by the Old Standard Birmingham Wire Gauge, the equivalent in inches being:

000	.425	2	.284	6	.203	10	.134	14	.083
00	.380	3	.259	7	.180	11	.120	15	.072
0	.340	4	.238	8	.165	12	.109	16	.065
1	.300	5	.220	9	.148	13	.095	17	.058
								18	.049

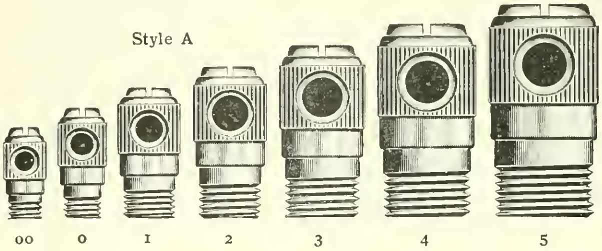
See also preceding page.

W. D. & C. F. TUCKER

HARTFORD, CONN.

LUBRICATING DEVICES

TUCKER'S TWO-PIECE OIL HOLE COVERS AND CUPS

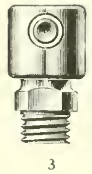


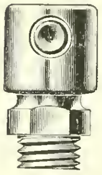
Special Lengths and Threads to Order

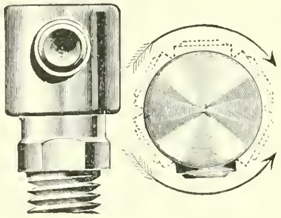
Made like a valve. Oil tight. Largest reservoir, size for size, of any oil hole covers on the market. Superior in style, finish, utility and uniformity. Samples are our selling agents. Prices subject to discount.

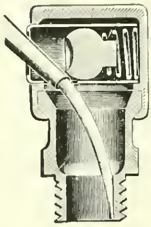
	Size of Body	Size of Thread	Diameter of Head	Weight Per 100	Price Nickel Plated Per 100
No. 00.	3/16	5/32 x 40	7/32	1/4 lbs.	$ 7.00
No. 0.	7/32	10 x 32	1/4	1/2 lbs.	7.00
No. 1.	1/4	1/4 x 32	5/16	3/4 lbs.	7.00
No. 2.	5/16	5/16 x 32	3/8	1 1/4 lbs.	9.50
No. 3.	3/8	3/8 x 24	7/16	1 7/8 lbs.	10.70
No. 4.	7/16	7/16 x 24	1/2	2 3/4 lbs.	12.50
No. 4.	7/16	1/8 in. pipe	1/2	3 lbs.	12.50
No. 5.	1/2	1/2 x 24	9/16	4 lbs.	16.00
No. 5.	17/32	1/4 in. pipe	5/8	4 1/2 lbs.	18.00

SELF-CLOSING ROTARY HEAD—STYLE B.









	Size of Head	Size of Thread	Weight Per 100	Nickel Plated Price Per 100
No. 3.	7/16 in.	1/4 in. 32 pi.	1 3/4 lbs.	$12.50
No. 4.	1/2 in.	5/16 in. 32 pi.	2 1/2 lbs.	15.00
No. 5.	5/8 in.	3/8 in. 24 pi.	4 3/4 lbs.	18.00
No. 5.	5/8 in.	1/8 in. pipe	5 lbs.	18.00

An Oil Hole Cover with self-closing port on the side. Head rotates, placing port in handiest position to lubricate. A cup designed for the operator that never shuts a door or pulls the plug in a wash basin costs more than style A on this account. Taps for all sizes carried in stock.

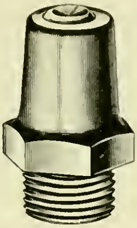
Sample and discount mailed on application.

Send for Blue Print Dimension Chart

W. D. & C. F. TUCKER

HARTFORD, CONN.

TUCKER'S SELF-CLOSING TOP OILERS—STYLE C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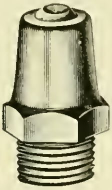
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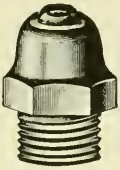
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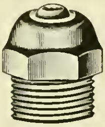
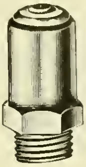
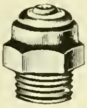

3 Special


4

Special sizes and pitches of threads made to order from sketches, quantity making the price. Samples sent on request. Useful in unhandy places with snout of oil can.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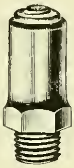
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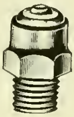
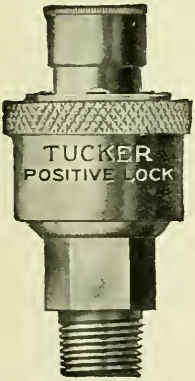

7

	Size of Thread	Size of Hexagon	Weight Per 100	Nickel Plated Per 100
No. 1.	¼ in. pipe	⅝ in.	4½ lbs.	$20.00
No. 1.	½ in. x 24 pi.	⅝ in.	4¼ lbs.	20.00
No. 2.	$\frac{7}{16}$ x 24 pi.	½ in.	2½ lbs.	15.00
No. 2.	⅛ in. pipe	½ in.	2½ lbs.	15.00
No. 3.	$\frac{7}{16}$ in. 24 pi.	½ in.	2½ lbs.	12.00
No. 3.	⅛ in. pipe	½ in.	2¼ lbs.	12.00
No. 3. spec.	⅛ in. pipe	$\frac{7}{16}$ in.	1¾ lbs.	11.00
No. 4.	¼ in. 32 pi.	⅜ in.	1½ lbs.	8.00
No. 4.	$\frac{5}{16}$ in. 32 pi.	⅜ in.	1½ lbs.	8.00
No. 5.	$\frac{5}{16}$ in. 32 pi.	⅜ in.	⅞ lbs.	6.00
No. 6.	¼ in. 32 pi.	$\frac{5}{16}$ in.	1 lbs.	7.00
No. 7.	¼ in. 32 pi.	$\frac{5}{16}$ in.	¾ lbs.	5.50

TUCKER'S POSITIVE LOCK COMPRESSION GREASE CUP



Patented Jan. 11, 1910

Fitted with a barrel lock. No springs and washers to depress when feeding the lubricant. Made to gage limits of .002 inch on fit of thread.

List Prices subject to Discount

No.	000	00	0
Inside Diameter, Inches	¾	1	1¼
Shank Pipe Thread, Inches	⅛	⅛	¼
Grease Capacity, Ounces	¼	½	⅔
Plain Brass, each	$0.90	$1.10	$1.35
Polished, each	1.05	1.25	1.50
Nickel Plated, each	1.20	1.40	1.70

Cap member has internally milled threads. Base member sized with hand die. We are prepared to furnish the cap member from pressed brass with milled threads in three sizes—¾″-000, 1″-00, 0-1¼″. All with 24 pitch thread, U.S. form.

Send for Blue Print Dimension Chart

AMERICAN GAS FURNACE COMPANY

Established 1879—Incorporated 1887

24 JOHN STREET - - NEW YORK

GAS ENGINEERS AND MANUFACTURERS

SPECIALTIES:

THE AUTOMATIC HEAT CONTROLLER

Positively regulates heat with no greater variation than 5° Fahrenheit.

GAS BLAST FURNACES

For all Industrial Heating Processes.

HEATING MACHINES

For the automatic transmission of work through properly heated space.

BURNERS AND BLOW PIPES

Of great variety and novel design.

CASE HARDENING MACHINES

Carbonizing by gas instead of solid carbon.

ROTATING RETORTS, Sealed

For Annealing, Hardening, Tempering or Coloring steel work.

ROTARY PRESSURE BLOWERS

Indispensable for proper Gas Combustion.

CATALOGUES AND DESCRIPTIVE PAMPHLETS

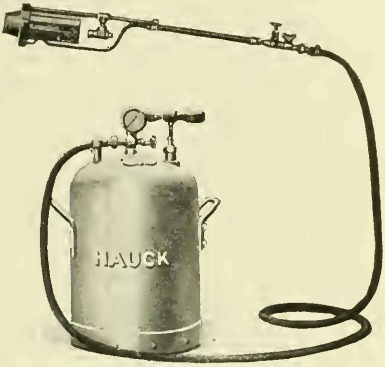
On application.

THE USES OF GAS in Mechanical Heating Processes are only **BEGINNING TO BE APPRECIATED. IT PAYS TO CONSULT US,** who are the Pioneers in that line.

HAUCK MANUFACTURING COMPANY

140 Livingston St., BROOKLYN, N. Y.

"HAUCK" PATENT OIL FUEL BURNERS AND KEROSENE TORCHES



Independent Self Contained Type

THIS CUT SHOWS HAUCK PATENT INDEPENDENT SELF-CONTAINED TYPE OIL BURNER OUTFIT, FOR KEROSENE OIL ONLY. This outfit consists of the seamless steel tank, equipped with hand air pump, gauge, all the fittings, 12′ oil hose, and the patented burner, with regulating valves, special oil needle valve strainer, all complete, ready for use.

This outfit is designed for field and shop work, where compressed air is not available or convenient for use. Flame is easily regulated and always under the control of the operator. Burner operates approximately for two hours, with single pumping of about five (5) minutes.

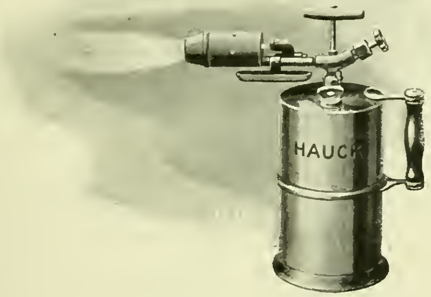
Hauck patent compressed air style outfits are made for use under any air pressure up to 100 pounds or more, with any grade fuel or crude oil.

HAUCK PATENT PORTABLE OIL FUEL BURNERS are invaluable for service in *Boiler Shops, Machine Shops, Coppersmith Shops, Foundries, Forge Shops, Railway Shops, Shipyards, and for structural work.*

They produce an intense heat which can be applied at any desired point without unnecessarily heating adjacent parts. For welding or straightening broken or bent locomotive frames, straightening out damaged steel platework, and for preheating preparatory to any welding process, their worth has been proved by the largest manufacturing firms in the country.

These burners are the most powerful on the market and absolutely reliable.

The price of a burner may often be saved on a single job.



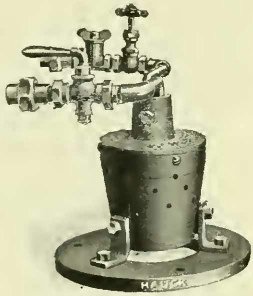
KEROSENE TORCH

For light brazing, wiping joints and similar heating operations, this portable blow torch is very efficient, producing an intense clear flame.

It is more suitable for general shop work than the dangerous gasolene torch.

Used largely in foundries and machine shops for drying out moulds, etc.

One Hauck Torch will equal the work of four gasolene torches.



SPECIAL FURNACE BURNERS

This illustration shows a Hauck Furnace Burner operated with compressed air or steam, and any grade of fuel oil. Especially suitable for attaching to boilers and furnaces.

We guarantee Hauck Burners to use less fuel and to give better results than any other make on the market.

Complete Catalogs on request.

THE FALK COMPANY

MILWAUKEE, WISCONSIN

MANUFACTURERS OF PRECISION HERRINGBONE GEARS WITH STAGGERED TEETH

(Wuest Patents)

We manufacture a complete interchangeable system of herringbone gears, with teeth generated on special machines, designed and built exclusively for our own use.

The gears which we produce are hobbed, both sides at once, in solid blanks.

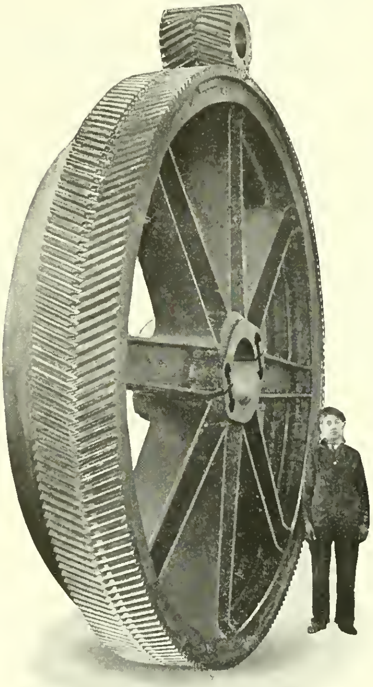


Fig. 1

The Wuest System of staggered teeth, besides giving the maximum contact surface for a given width of face, is invaluable in securing unbroken continuity of engagement when using high ratio pinions with very few teeth.

Other distinctive features:—

Highest attainable accuracy.

Involute tooth form on *circumferential* section.

Invariable spiral angle.

Perfect interchangeability.

Equal efficiency in both directions.

Wuest herringbone gears transmit power by smooth, continuous action without jar, shock or vibration.

They are almost noiseless.

They can be used for extremely high *single* gear ratios. In this connection we make a specialty of forged pinions in one piece with their shafts. Ratios of 15 to 1 are quite normal and 20 to 1 may be used when necessary. Wuest gears can be run with safety at far higher velocities than the spur type. Special gears for use in connection with steam turbines are suitable for speeds up to 7000 feet per minute.

Referring to the illustrations, Figure One shows a combined Steel Gear and Clutch Drum for double drum hoist at a Michigan Iron Mine.

The steel gear rings and solid forged pinions in Figure Two represent a type of high ratio drive adapted for use with conical and straight tube mills.

Figure Three shows the type used for steam turbine and high velocity work, with fine pitch and relatively wide face.

THE FALK COMPANY

MILWAUKEE, WISCONSIN

MANUFACTURERS OF PRECISION HERRINGBONE GEARS WITH STAGGERED TEETH

(Wuest Patents)

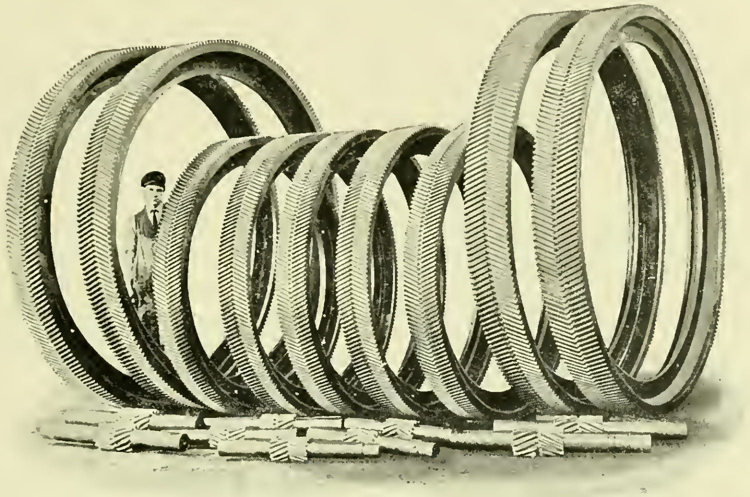


Fig. 2

SPECIAL ADVANTAGES

Long life.

High efficiency (loss never exceeds 1% at rated load).

Elimination of countershafts and double-gear trains.

Absence of vibration with prevention of shaft-crystallization and breakdown of motor insulation.

Quiet action with durable steel pinions.

The range of application for Wuest herringbone gears covers every case where spur gears are used and many new fields where spur gears are impossible.

Specially adapted for

- Marine Steam Turbines.
- Turbo-Generators.
- Turbine-driven centrifugal pumps, mills and shafting.
- Rolling Mills and Rod Mills.
- Tube Mills and Crushing Plant.
- Power Pumps.
- Air Compressors and Blowers.
- Hoisting, Elevating and Conveying Plant.
- Rubber Machinery.
- Machine Tools.

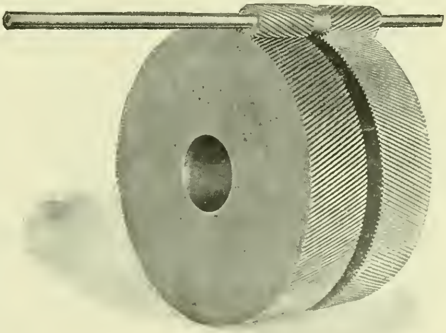


Fig. 3

J. H. WILLIAMS & CO.

BROOKLYN, N. Y.

SECURE THE 1912 HAND BOOK OF

SUPERIOR DROP-FORGINGS

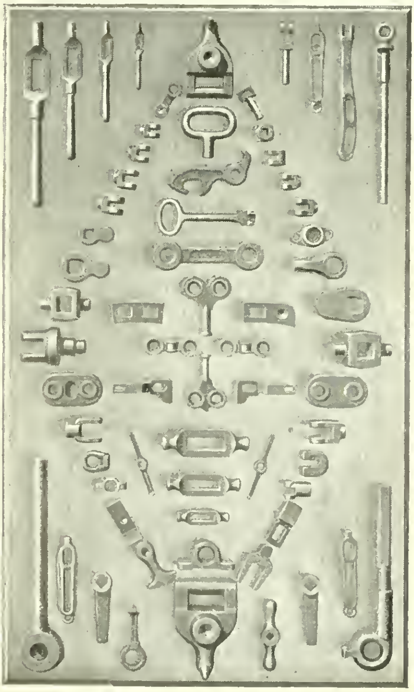
The pocket size book will not only show you the latest standard prices of our many drop-forged stock tools and accessories, but should convince you of our ability to take care of such "made to order" pieces as you may require in connection with your business affiliations.

It will prove a handy reference
book to you for anything in
Superior Drop-forgings!

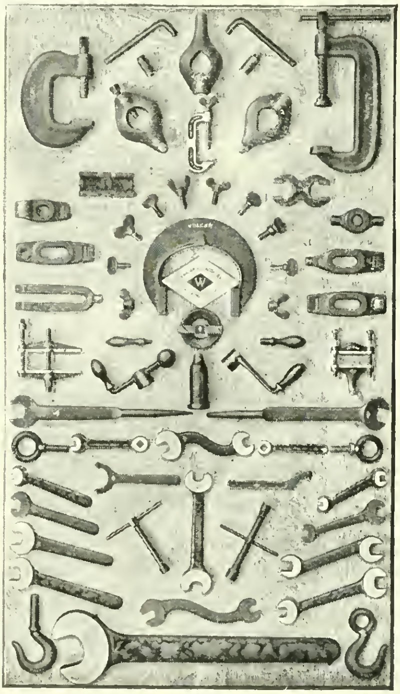
Among the new lines of Drop-forged tools you will notice that the "Vulcan" Safety Lathe Dogs are so constructed that all danger from sleeve contact has been eliminated.

The new "Agrippa" Clamps present unusual features at same price customarily paid for the cast steel product. The drop-forged "Agrippa" Clamps are lighter, stronger, have greater utility and will accomplish everything possible with the heaviest form of cast steel product.

Moreover, you will find interest in a number of other tools which possess points of merit and convenience unique to themselves; secure the book!



Special Forgings



Stock Forgings

STEEL CAR FORGE COMPANY

FORGING SPECIALISTS

General Offices

FRICK BUILDING, PITTSBURG, PA.

Works

ELLWOOD CITY, PA. **BUTLER, PA.** **HAMMOND, IND.**

Branch Offices

170 Broadway, New York City

1608 Fisher Building, Chicago, Ill.

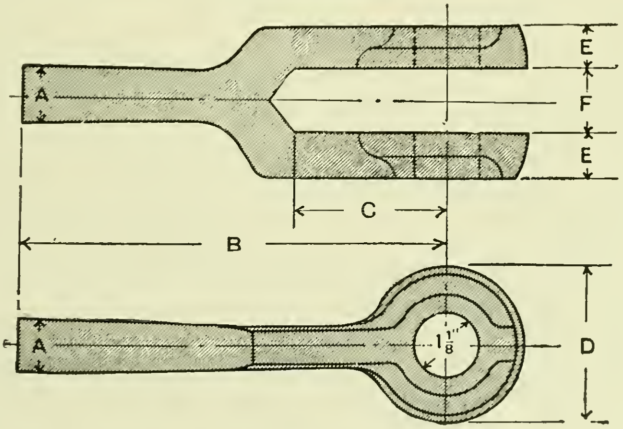
Representatives

F. A. Barbey, Boston, Mass.
R. H. Pilson, Washington, D. C.
Rank & Goodell, St. Paul, Minn.
R. R. Hammond & Company, Phila., Pa.
Adreon Manufacturing Co., St. Louis, Mo.
H. D. Cushman Co., Cleveland, Ohio.
Pierson, Roeding & Co., Pacific Coast

We manufacture a full line of iron and steel forgings for standard work, such as freight cars—also for special purposes where castings do not give sufficient strength, having designed many forged parts replacing castings.

Our equipment enables us to furnish to advantage drop forgings, bulldozer or bending work, upset forgings, welding, punching, shearing, eye bending, hooks, etc.

Drop Forged Jaw



Correspondence solicited

THE AMERICAN BRASS COMPANY
THE COE BRASS BRANCH

ANSONIA, CONN

EXTRUDED METALS

The word "extrusion" accurately describes the operation by which the various sections are formed.

A billet of brass is first cast of a convenient size and suitable composition for the purpose intended. This billet is re-heated until it is of a plastic consistency, and then, by means of a small crane, it is placed within a very strong horizontal cylinder or "container," at the front end of which is the die. Upon hydraulic pressure being applied at the rear end of the container the plastic metal is forced or squirted through the die, issuing therefrom in a long bar having a cross-section corresponding to the hole pierced in the die.

The very high pressure (oftentimes as high as 60,000 pounds to the square inch) to which the semi-plastic metal is thus subjected gives it increased density and renders it perfectly homogenous and free from possible casting defects. The bars produced by this process have a greater strength and tenacity than those made by rolling or other cold working methods, and for special purposes alloys are made having the requisite strength and elongation for work demanding the strength of good quality steel. It is also obvious that sections which cannot possibly be rolled or drawn can be made by the Extrusion Process with great accuracy.

In addition to bars or rods of irregular cross section, all stock sizes of round, hexagon, square, rectangular, half-round, etc., are produced suitable for forging and all engineering purposes. Alloys used for this purpose are Nickel Bronze, Muntz Metal, and other special bronzes suitable for forging. Free cutting brass rods are made also for manufacturing purposes. The alloys used for making angles, channels, T-bars and special shapes for marine work have a high tensile strength and are particularly non-corrosive in their qualities.

A considerable use that has been found for this metal is in the form of mouldings for architectural work. It is well adapted for use in large stores and office buildings, railroad stations, residences, etc., and can be oxidized to give it any color desired.

Extruded mouldings are also supplied for use in the construction of passenger and sleeping cars, as the metal will take a high finish, is durable, and does not rust out or corrode in service. It is also largely used for step treads, step nosings, platform bindings, hinges, and door stops, having taken the place of brass castings for these purposes. In fact extruded shapes made in a free drilling and free milling stock have superseded castings for many purposes. The metal being perfectly smooth and accurate requires little if any machining, and is utilized in a great variety of manufacturing lines, small pieces being sawed or milled from special shape extruded brass bars to take the place of machined castings. This affords an efficient and economical means for making small, intricate parts such as cams, dogs, pinion wheels, and various other parts that are required in the manufacture of locks, typewriters, telephone and telegraph instruments, switchboards and other electrical apparatus.

The dimensions of the finished bars are limited by the weight of the billets used, and range in weight from 100 to 140 pounds. We are able to produce special shape bars or rods ranging from light sections of about 3-8 in. diameter and weighing a fraction of a pound per foot, to heavy sections 5 or 6 inches in width and weighing many pounds per foot. It will thus be understood that light sections may be had in long lengths and heavy angles, etc., in lengths of from 12 to 18 feet.

DOEHLER DIE-CASTING COMPANY

BROOKLYN, N. Y., U. S. A.

DIE-CASTINGS.

The accompanying cuts are photographic reproductions showing the sharp outlines and smooth finish of our castings. These die-castings are made in steel dies by our patented process, and represent the modern economical way of producing finished parts rather than by machining.

Parts heretofore considered machine impossibilities become, by our process, realities. Inventions shelved as impractical or too costly to produce become money makers.

Cheaper production on intricate parts is not the only saving afforded by this process. Die-cast parts are exactly alike, and absolutely accurate, so that time gained in the assembling room is an important factor.

It is a foregone conclusion, considering the expense of a steel-die, that parts wanted in small quantities usually are not die-casting propositions. Cast-iron parts with little machining do not prove any saving, the handicap on the cost of material is too great to overcome to make an attractive showing.

Generally speaking, intricate parts made in brass and used in large quantities yield the greatest saving. However, circumstances and individual cases differ so much that no general rule can be outlined and, therefore, we gladly give our advice on any part at any time, or enter into correspondence on all subjects relating to die-casting with parties interested or seeking information.

The following is a brief attempt to give the main characteristics of our different alloys and their usefulness.

"H" metal. Main ingredient: zinc.

Very hard, wears splendidly as a bearing metal, somewhat brittle against a blow; compares favorably with cast-iron. Will plate in any color.

"T" metal. Main ingredient: zinc.

Alloyed with tin and copper, producing a pliable, tough metal; compares favorably with brass. Will plate in any color.

"L" metal. Main ingredient: lead.

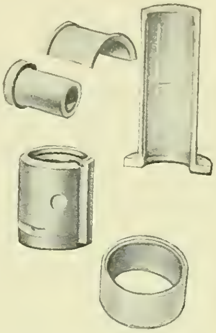
Hardened by tin and antimony. Well adapted for parts to resist steam and water.

"A," "A" metal. Main ingredient: tin.

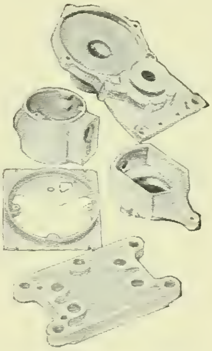
A good, low-priced, non-corrosive, anti-friction babbitt.

"G," "B" metal. Main ingredient: tin.

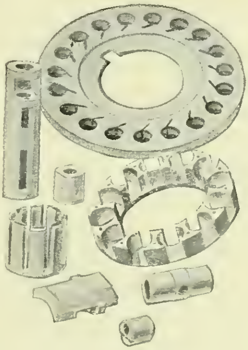
A high-class tin alloy, comparatively hard, non-corrosive; produces castings of high finish and sharpness: wears splendidly in bearings for gas engines. This alloy gives great satisfaction on automobile engine bearings.



Die-Cast Babbitt Bearings and Bushings.



Automobile Accessories.



Ball and Roller Bearing Retainers.

THE CARBORUNDUM COMPANY

NIAGARA FALLS, N. Y.

New York, Chicago, Boston, Pittsburg, Philadelphia, Cleveland, Cincinnati, Milwaukee,
London, Eng.

MANUFACTURERS OF ABRASIVE MATERIALS, CARBORUNDUM AND ALOXITE GRINDING WHEELS, CARBORUNDUM PAPER AND CLOTH, ALOXITE CLOTH, CARBORUNDUM BRAND GARNET PAPER, CARBORUNDUM SHARPENING STONES

CARBORUNDUM GRINDING WHEELS

FOR cast iron, brass, bronze, aluminum, general machine shop work, for cylindrical, internal or surface grinding of metals of low tensile strength, for grinding pearl, marble, granite and porcelain.

ALOXITE GRINDING WHEELS

FOR steel grinding, malleable iron, reamers, cutters, drills, planer tools, knife grinding, cylindrical, internal or surface grinding where the material is steel.

Made in any standard shapes or sizes, or in any special shapes or sizes, subject to blue prints.

THE RIGHT WHEEL IN THE RIGHT PLACE

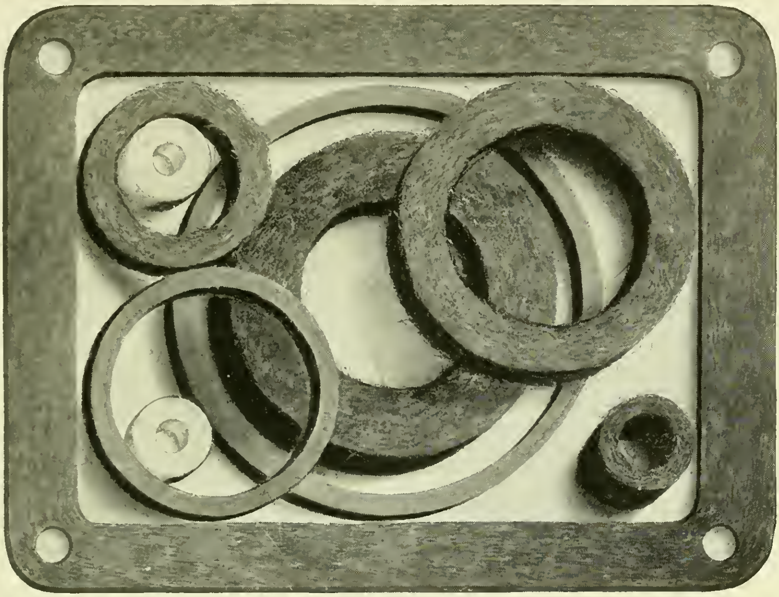
This is the secret of efficient and economical grinding. Our service department is at your command to give you the benefit of years of experience in all classes of grinding—to give you proper wheel-right grit, right grade. Let us know about your grinding conditions—there is a Carborundum or Aloxite wheel to meet every grinding condition.

Performances—things actually done by Carborundum and Aloxite wheels—are the most convincing arguments in favor of their merits. Send for these illustrated records—they are shown in our "efficiency booklet." Let us arrange to send you a trial wheel. We can better your grinding conditions.

N. E. BOOTH

644 PACIFIC STREET, BROOKLYN, N. Y.

MECHANICAL FELT GOODS OF ALL KINDS



BOOTH FELT WASHERS, GASKETS, PACKINGS, OIL RINGS, LUBRICATING PADS, ETC.

Booth is a specialist in felts and in the cutting of felts for commercial purposes. He has a three-story factory full of assorted felts, dies, machines, and expert workmen, *and he has a system.* He generally ships on the day after receiving an order,—and he knows how to cut costs as well as felts.

Time was when the user made these articles himself. But it didn't pay. It took too long to get the felt. It cost too much to get the dies. Machines that were built for something else made awkward business of the work.

So Booth stepped in. He designs his own machines. He has shelf upon shelf of interchangeable dies, all made by his own men. The line of round washer dies, for instance, includes every size from ⅛ inch to 24 inches. Once in a while some one comes at Booth with an order for felt cut into some freak shape. He hasn't the dies made up, but he has the facilities for making them in a hurry, and the bill is rendered only for material and work. It does not include the cost of dies.

Booth makes a specialty of supplying any of these goods in very small quantities for experimental or test purposes.

Booth saves time and money every day for a lot of big manufacturers in a good many lines of business, and can do the same for you. Ask him about it.

ALLITH-PROUTY COMPANY

GENERAL OFFICES AND FACTORY

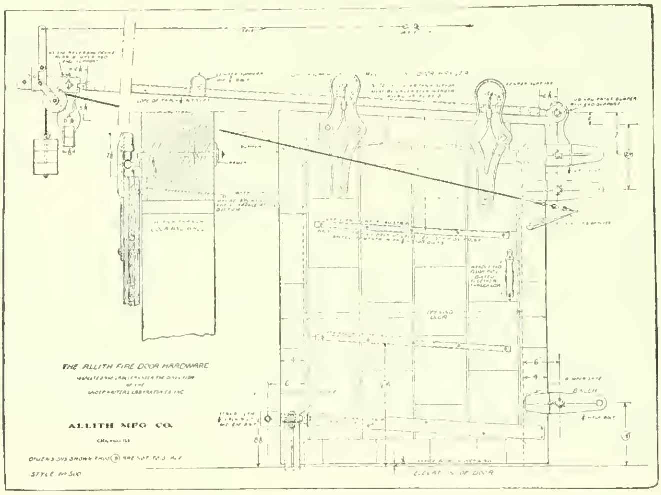
DANVILLE, ILLINOIS

BRANCH OFFICES AND WAREHOUSES

Chicago	New York	Philadelphia
Boston	San Francisco	Los Angeles

Manufacturers of

DOOR HANGERS FIRE DOOR HARDWARE BUILDERS' HARDWARE
STORE LADDERS MERCHANDISE CARRIERS FLOOR HINGES



Allith hardware is inspected by the Underwriters' Laboratories Incorporated, under direction of the National Board of Fire Underwriters.

We manufacture the largest and most complete line of Sliding Door Hardware suitable for carrying sliding doors of all kinds, and call special attention to our round track hangers for warehouse and factory doors of different styles, weights and thicknesses.

Allith Fire Door Hardware is the easiest and simplest to install, strongest and most efficient, durable and satisfactory hardware made. Automatic closing devices are supplied in various styles to meet any requirements, none of them interfering with the free and easy operation of doors.

Our Parallel Door Equipment insures perfect movement of continuous parallel doors, permitting an opening at any place desired, and is the only absolutely storm-proof arrangement.

Our Merchandise Carriers for factories and warehouses are the simplest, strongest and easiest-running. Made from the highest grade malleable iron and guaranteed in their various sizes to carry safely any loads from 250 pounds to 4000 pounds.

Catalogues, detail drawings, specific information and estimates gladly furnished on request.

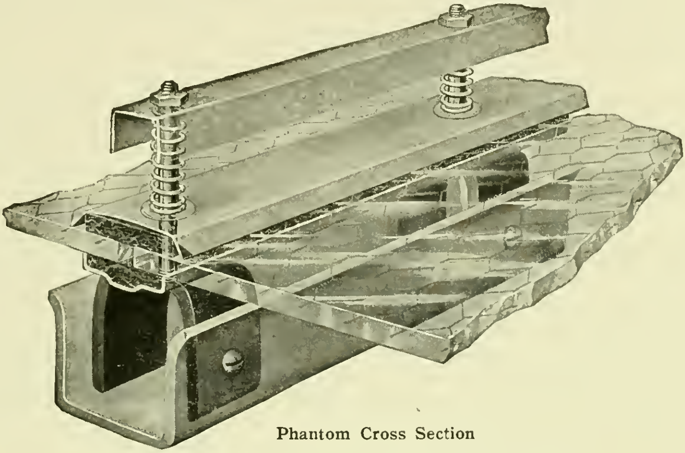
THE G. DROUVÉ COMPANY

BRIDGEPORT, CONN.

Telephone Connections
Western Union Code

Branch Offices
180 N. Dearborn St., Chicago, Ill.

"ANTI-PLUVIUS" PUTTYLESS SKYLIGHT AND SASH OPERATING DEVICES



Phantom Cross Section

"ANTI-PLUVIUS" PUTTYLESS SKYLIGHT

The "Anti-Pluvius" Skylight is weather-proof. It can be furnished in any type and will be found to be a permanent construction. The glass is bedded on cow-hair felt which provides a cushion resting surface to take up shock, vibration or expansion and contraction. The weight of a man on the bridge is carried through to the supporting channels below without contact with or pressure on the glass. Each light is independent of every other and does not come in contact with metal, thus doing away with condensation from sweating. Manufacturers in general are gradually replacing old and worn out skylights with the "Anti-Pluvius," thereby establishing a standard and doing away with much crackage and breakage of glass. Other information furnished on request, together with estimates.

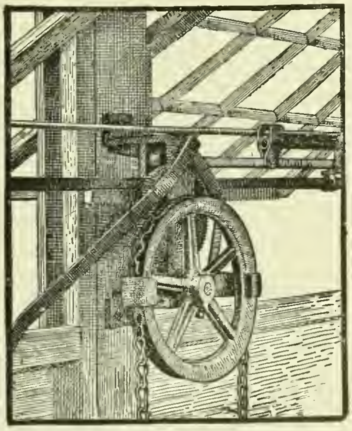
"STRAIGHT-PUSH" SASH OPERATOR

The "Straight-Push" Sash Operator operates any number of sash positively, quickly, and with individual power applied to each sash. A rack and pinion moving a line of $\frac{3}{4}''$ pipe, supported by brackets, backwards and forwards, is the guiding principle, and a push or pull is secured instead of torsion.

The main guide lever-arms are of $\frac{1}{4}''$ steel. The supporting brackets are formed of $\frac{1}{4}''$ steel with cast-iron spools on bearing shafts of phosphor bronze. The operating wheel has cut-steel gears with steel shaft controlling a cut-steel rack and pinion. The few connections are made with phosphor bronze washers between to prevent these parts rusting together.

A one-man control gives sufficient power to operate lines of sash 100, 200 or more lineal feet.

Full information on request.



Straight-Push Sash Operator

THE BRAEBURN STEEL COMPANY

BRAEBURN, PENNSYLVANIA

"THE TOOL STEEL MILL"

GRADES 1, 2, 3

These grades are carbon crucible steels corresponding in properties respectively to the best makes of standard, extra, and special tool steel. They are made in all tempers and each grade is uniform both as to chemical analysis and method of manufacture as well as to working properties. A reference to our catalogue will give more information regarding these steels.

GRADE 4

This is a special steel for Turning, Planing and Slotting hard materials, Expensive Cutters, Drills, Forming Tools, Taps, Punches, Dies, etc., and is particularly recommended for all kinds of tools where special endurance is sought and where great strength and toughness are required. This steel is made in all tempers.

GRADE 5

This is a very hard steel made in one temper only for Turning, Planing, Slotting and working Chilled Iron or any other hard material and especially for those purposes where the tool has to carry a fine edge and has to put a very fine finish on the work. It is not intended for tools which have to withstand shocks like the blows of a hammer or sledge.

Treatment. This steel should be given more time in heating for forging and hardening than other tempering steels. For forging, heat slowly and uniformly to a bright red, then forge the tool, using light blows as the heat dies out; do not hammer at a black heat. For hardening, reheat to a dark red and quench in warm water.

Use a wet grindstone in grinding tools made from this steel.

GRADE 6

This is an oil hardening steel, made only in one temper, for Milling Cutters, Taps, Reamers, Gauges, Hard Steel Bushes, Ball Bearings and other purposes where it is particularly desired there shall be neither contraction nor expansion after hardening.

Treatment. For forging, heat slowly and uniformly to a bright red, harden at a dull red, about 1350 degrees Fahrenheit and draw the temper only sufficiently to relieve the strain. Hardened in this manner it is very tough and takes an excellent cutting edge and practically obviates all contraction and expansion.

GRADE 7

This is a deep hardening steel and is especially recommended for Twist Drills and is made in all tempers.

SELF HARDENING

This is a special alloy steel which cannot be cut or punched cold but can be shaped or ground on a stone or emery wheel. It is suitable for moderately high speeds where great strength is not necessary.

HIGH SPEED STEELS

Made in two grades known as "High Speed" and "High Speed Special."

The HIGH SPEED STEEL for Punches, Boring Tools, Straight Drills, Twist Drills, Milling Cutters, Gear Cutters, etc., is capable of doing all the work of ordinary machine shop equipment with heavy cuts and coarse feed at either high or low speed.

The HIGH SPEED SPECIAL STEEL is for purposes similar to those for which the High Speed Steel is made but is adapted more particularly to cases where the service is unusually severe. We recommend the use of this steel only where unusual service is demanded or where it is desirable to use a tool for a considerable length of time without re-grinding. This steel, owing to its superior quality and toughness, of course holds its edge longer and stands greater strains than does the High Speed Steel, but its use is hardly necessary or economical except under unusually severe conditions.

B. T. GRADE

A common tool steel suitable for Springs, Shafts, Forgings, Hammers, Picks, Forks, Rakes, Hoes, Corn Stalk Cutters, Cutlery, Lawn Mowers, Harvesting Machinery, Wedges, Swedges, Die Blocks, Cant-Hooks, Files, etc., or wherever well selected and carefully worked steel is used.

THE COLONIAL STEEL COMPANY

PITTSBURGH, PENNSYLVANIA

HIGH GRADE STEEL

COLONIAL HIGH SPEED STEEL

Adapted for the heaviest cuts or the highest speeds on all classes of material. Colonial High Speed Steel is made in one grade only and is capable of doing any class of work for which high speed steel is suited.

COLONIAL BEST TOOL STEEL, WATER HARDENING

An alloy steel adapted for fine finishing cuts, such as forming tools, cutters, hobs, threading dies, etc.; for brass and copper lathe tools, and other purposes where machines are not equipped to use high speed tool steel.

COLONIAL NO. 7 TOOL STEEL, WATER HARDENING

A vanadium tool steel of great strength and toughness; made from pure wrought iron, and suitable for high-class tools of all kinds, especially those subject to strain or stress of any kind through repeated action or repeated shock.

COLONIAL SPECIAL TOOL STEEL, WATER HARDENING

A straight carbon tool steel, made from pure melting iron. Suitable for shop tools of all kinds.

RED STAR TOOL STEEL

Standard grade for ordinary purposes.

RED STAR DRILL STEEL

For rock drilling purposes.

NICKEL STEEL

Bars and Billets made in small furnaces and carefully melted to insure homogeneous steel. Furnished free from pipes, seams and all defects.

30 PER CENT NICKEL STEEL

Anti-corrosive. Used for valve stems and parts of internal combustion engines. or other purposes where material is desired that will not rust or corrode.

SHEET STEEL

For knives and tools of all kinds, springs, agricultural implements, etc.

SOFT CENTER PLOW STEEL

Made in slabs or sheared to pattern, carefully manufactured, and special attention given to the toughness of center, and rigidly inspected.

FIVE PLY JAIL BARS AND SAFE PLATES

Send for Catalog

HALCOMB STEEL COMPANY

SYRACUSE, N. Y.

BRANCHES, AGENCIES AND STOCKS:

Halcomb Steel Co., Syracuse, N. Y.
Halcomb Steel Co., Cleveland, O.
Halcomb Steel Co., Philadelphia, Pa.
Halcomb Steel Co., Chicago, Ill.
Hawkridge Bros. Co., Agents, Boston, Mass.

HIGH SPEED STEEL, TOOL STEEL, SHEET STEEL, DRILL RODS, FINE WIRE, COLD DRAWN STEEL AND SPECIAL STEELS OF ALL KINDS

HALCOMB SPECIAL HIGH SPEED TOOL STEEL

This steel is unsurpassed by any product, foreign or domestic, where the most severe duty is demanded, as in heavy lathe and planer work, boring tools, turning glazed tires, alloy steel gears, flat and twisted drills. It is noted for consistent uniformity of analysis, and does not require fussy treatment.

The risk of breaking in hardening is minimized and tools cannot be ruined by over-heating. These steels are also readily annealed for machining. Cutters and fragile tools do not require a sweating heat when made from our steels. They retain an excellent cutting edge, and hence show high efficiency for both roughing and finishing work.

HALCOMB KETOS OIL HARDENING TOOL STEEL

The greatest achievement in tool steel since the discovery of high speed steel. There is a place for this steel in every shop and tool room. This steel neither expands, contracts nor warps in hardening. Safest steel ever produced from which to manufacture intricate tools, dies, cutters, etc. Think of making a twenty-two inch tap with no change in pitch after hardening! Write for special booklet on Ketos.

HALCOMB SPECIAL TOOL STEEL

This steel is made in six tempers (varying from high to low carbon), to meet all tool steel requirements.

For turning, planing, and slotting hard materials, for expensive cutters, drills, forming tools, taps, reamers, punches, dies, etc.

Particularly recommended for all kinds of tools where special endurance is sought and where great strength and toughness are required.

HALCOMB EXTRA WARRANTED TOOL STEEL

This steel is made in six tempers to meet varied requirements, and is of fine quality, suitable for many kinds of tools, including wood-working bits and knives, cold chisels, drills, mining drills, granite tools, lathe and planer tools, cutters, taps, reamers, dies, shear blades, punches, etc. An excellent steel for general use.

OTHER TOOL STEELS

Other well-known brands of tool steels manufactured by this company are:

"Air Hardening Tool Steel"
"Double Special Tool Steel"
"Double Extra Tool Steel"
"Extra Special Tool Steel"
"Peerless Tool Steel"
"Standard Tool Steel"

Catalogues on request

HALCOMB STEEL COMPANY

PIONEER AMERICAN MAKERS OF ELECTRIC FURNACE ALLOY STEELS

These steels are chemically pure and free from sulphur, phosphorus and gases. They are completely deoxidized, thoroughly melted, homogeneous, and free from segregation, seams, and surface defects. They are easy to machine and respond with certainty to any definite heat treatment. They are very readily drop forged, and are not apt to be injured by over-heating.

Electric Furnace alloy steels have wide hardening ranges, so that uniform results can be had in plants not equipped with the latest devices for controlling temperatures. In resistance to fatigue they stand from 50% to 100% better than corresponding compositions made by other processes.

We make a specialty of Alloy Spring Steels, which are characterized by greater strength and fatigue values than ordinary spring steels. They thus permit of greater fibre stresses without sacrificing present factors of safety, and consequently make possible substantial reductions in weight.

These alloy steels are especially well suited for the construction of machine tool and automobile parts.

ELECTRIC NICKEL STEEL

Nickel Steel is perhaps the most generally useful of the alloys. In various conditions of thermal treatment, it displays wonderfully good qualities, and is well suited to the production of structural parts. It case-hardens well, and in machining qualities is superior to the usual alloy steels.

CHROME VANADIUM STEELS

The manufacture of genuine Crucible or Electric Chrome Vanadium Steels constitutes the highest attainment of the steel makers' art. By the proper blending of these alloying metals and adjustment of the carbon, we not only obtain every static property that can be obtained from Nickel, Chrome Nickel, Silico-Manganese or other alloys, but also obtain wonderfully enhanced dynamic or antifatigue qualities.

In general, when a better material than Nickel Steel is needed, we unreservedly commend our Chrome Vanadium Steels.

OTHER ELECTRIC FURNACE ALLOY STEELS

The Halcomb Steel Company also manufacture Chrome-Nickel Steel, Chrome-Silicon Steel, Silico-Manganese Steel, and other alloys to meet special requirements.

COLD DRAWN AND COLD ROLLED STEEL

We are prepared to furnish cold drawn steel from .007 in. to 3 in. diameter in rounds and corresponding sizes in squares, flats and special shapes, exact to size.

HEAT TREATING AND ANNEALING

We are prepared to furnish any of our products, either tool or alloy steels, hammered or rolled, in the form of billets, bars or forgings, and unannealed, annealed or oil-tempered. Our heat-treating facilities are unexcelled, thoroughly modern and up-to-date in every way and under competent technical supervision. We act in an advisory capacity for our customers in reference to all matters pertaining to heat-treatment and heat-treating equipment, pyrometers etc.

PARKESBURG IRON COMPANY

PARKESBURG, PENNSYLVANIA

1023-30 Church St., New York
1613 Fisher Bldg., Chicago
822 Arcade Bldg., Philadelphia

REPRESENTATIVES:

Arthur C. Harvey Co., Boston
Eagleston Parke, Inc., Norfolk
Woodward, Wight & Co., Ltd., New Orleans
H. A. Fuller Iron & Steel Co., St. Louis

CHARCOAL IRON BOILER TUBES, CHARCOAL IRON ARCH TUBES, CHARCOAL IRON SAFE ENDS, CHARCOAL IRON SUPERHEATER TUBES.

Our entire output consists of genuine old-fashioned charcoal iron, which resists pitting and corrosion, is free from crystallization, preserves its ductility and gives the lowest ultimate cost of boiler tube installation.

BOILER TUBES AND SUPERHEATER TUBES. Range of sizes, 1½″ O. D. to 6″ O. D.

Owing to the careful attention given all details, tubes produced by the Parkesburg Iron Company have remarkably few blisters, run true to gauge, and are of so soft and ductile a nature that they bead over the flue sheet with the greatest ease, and rarely require rerolling after installation.

For boiler and superheater tubes, charcoal iron is eminently suited. It is of a fibrous nature, and will not crystallize under shock or vibration. It is ductile, can be beaded easily and securely, and requires little calking. It is preeminently weldable. It is non-corrosive and will not pit.

ARCH TUBES.—Careful attention is given to arch pipe, which are furnished from the "Parkesburg Special" grade of iron identical with the former "Allison Special" which was favorably known for arch pipe in the past. As on the quality of arch pipe the lives of engineers and firemen are dependent, "Parkesburg Special" quality is recommended as being the safest on the market. The range of sizes of arch pipe and water grate is 1 7-8 inches to 4 inches, outside diameter, and from No. 10 B.W.G. to 5-16 inch thick.

SAFE ENDS.—The Parkesburg Iron Company has met with great success in safe ends both cut to length and in random lengths. These safe ends are furnished of charcoal iron and if required to American Master Mechanics Association Specifications, or of Solid Swedish Charcoal Iron, their advantage over steel being due to the fact that they can be welded to the body grade flue securely and permanently, thereby avoiding accidents to flues in service through failures at the weld.

SOME OF THE ADVANTAGES OF GENUINE CHARCOAL IRON ARE AS FOLLOWS:—It is not easily corroded or affected by electro-chemical action, being nearly pure iron. Because of its homogeneous structure, low manganese, and the intimate mixture of cinder surrounding each filament of iron it is not apt to become pitted.

It has unusually good welding qualities. It is very fibrous and ductile, and under test, shows uniformly distributed elongation. It is not subject to crystallization under vibration or shock.

Write for complete catalogue.

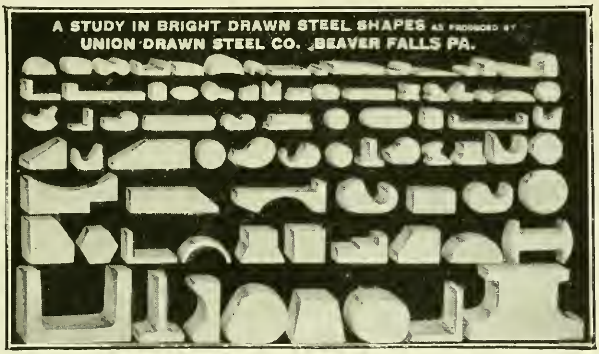
UNION DRAWN STEEL COMPANY

Works and General Office

BEAVER FALLS, PA.

Warehouses: New York, Philadelphia, Chicago, Cincinnati.
Branch Sales Offices: Boston, Buffalo, Atlanta.

MANUFACTURER OF BRIGHT FINISHED STEEL EXCLUSIVELY IN ROUNDS, SQUARES, HEXAGONS, FLATS AND SHAPES, SHAFTING, SCREW STEEL, AXLE STEEL, BESSEMER, OPEN HEARTH, CRUCIBLE, NICKEL AND VANADIUMS, DRAWN—COLD ROLLED AND TURNED STEEL.


A STUDY IN BRIGHT DRAWN STEEL SHAPES AS PRODUCED BY
UNION DRAWN STEEL CO. BEAVER FALLS PA.

SPECIAL SHAPES OF COLD DRAWN STEEL of any dimensions within our range and for all purposes, will be made in the shortest possible time consistent with perfection in quality, in accordance with specifications furnished, where sufficient quantity will justify equipment.

The most comprehensive stock of Bright and Finished Steel, Rounds, Squares, Hexagons and Flats carried at our branch warehouses, in addition to the large stock we carry at our mill.

We are the largest manufacturers of cold finished steel and iron for shafting and various machinery uses.

Established 1889, but rebuilt Fireproof Plant and all new machinery installed, 1911.

SHAFTING.—We use only the best quality of soft steel and are manufacturing under recent patents, covering machinery and appliances, by a process superior to anything known for producing work mathematically accurate as to size, absolute straightness, and a perfectly polished surface.

PISTON AND PUMP RODS.—For piston and pump rods we use a special grade of steel, and can produce them strictly uniform in size and quality, highly polished, perfectly straight, and of lengths up to 60 or 70 feet.

SCREW STEEL.—For this work we furnish a special analysis of steel, which, after years of experiment, has proved best adapted to free cutting and threading, and for the production of the maximum number of parts in the minimum of time, by the use of automatic and hand screw machines and Turret lathes.

SPECIAL STEEL.—For the various places where special grades of steel are required, our experience and facilities are such that we can promptly furnish material best adapted for the special requirements.

WHEELOCK, LOVEJOY & COMPANY

NEW YORK BOSTON

HIGH GRADE STEEL

Agents
THOMAS FIRTH & SONS, LTD.,
SHEFFIELD, ENG.
Cutlery and Saw Steel

FIRTH STERLING STEEL COMPANY,
PITTSBURG, PA.
Makers Highest Grade Tool Steel

WEST PENN STEEL CO.,
BRACKENRIDGE, PA.
Cold Rolled and Electrical Sheet Steel

Agents
GLOBE WIRE CO., LTD.,
SHARPSBURG, PA.
Polished Drill Rods, Needle Wire
Drawn Steel in Special Shapes

WEST LEECHBURG STEEL COMPANY,
PITTSBURG, PA.
Hot and Cold Rolled Strip Steel

BRIGHTMAN MFG. CO.,
COLUMBUS, OHIO
Turned, Ground and Polished
Shafting and Screw Stock

HY-TEN STEEL

This steel is of high tensile strength and elastic limit, especially intended for machine tool parts where good wearing qualities combined with great strength and toughness are essential.

A complete stock is carried in warehouses for prompt shipment.

FIRTH-STERLING "BLUE CHIP" HIGH SPEED STEEL

Suitable for Lathe and Planer Tools, Milling Cutters, Drills, Reamers, Taps, Cutting and Blanking Dies, etc.

"Blue Chip" High Speed Steel is carried in stock in the following sizes and shapes:

SQUARES, $\frac{1}{4}$ in. to $\frac{5}{8}$ in. Hard Steel ready for use.
$\frac{3}{16}$ in. to 3 in. Annealed.
ROUNDS, $\frac{3}{16}$ in. to 10 in. Annealed.
FLATS, $\frac{3}{8}$ in. x $\frac{1}{8}$ in. to $5\frac{1}{2}$ in. x $\frac{7}{8}$ in. Annealed.
$\frac{1}{2}$ in. x $\frac{1}{4}$ in. to $3\frac{1}{8}$ in. x 2 in. Annealed.

Special sizes can be secured promptly from the mill.

FIRTH-STERLING TOOL STEELS

Other high grade Firth-Sterling Steels carried in stock by Wheelock, Lovejoy and Company are in part as follows:

Firth's Best Tool Steel (Water Hardening), a strictly high grade carbon tool steel for general service.

Firth-Sterling Special Steel. For Punches, Dies, Chisels, Blacksmith Tools, Shear Blades, Rivet Snaps and all Shop Work.

Sterling Tool Steel. This steel is made to compete with the lower grades on the market, and will compare favorably with them. Carried in stock in Rounds, Flats, Squares and Octagons.

ALLOY STEELS

To meet the increased demand for steels that are more effective than carbon steels, and of a different character from High Speed Steel, we have developed the following which we now recommend for various purposes:

Firth-Sterling "Extra Special" Steel
Firth-Sterling "Double Special" Steel
"Hold Fast" Magnet Steel
"C Y W Choice" Steel
"A W Special" Steel
Firth-Sterling Finis Steel

"TOOL STEEL FOR EVERY PURPOSE"

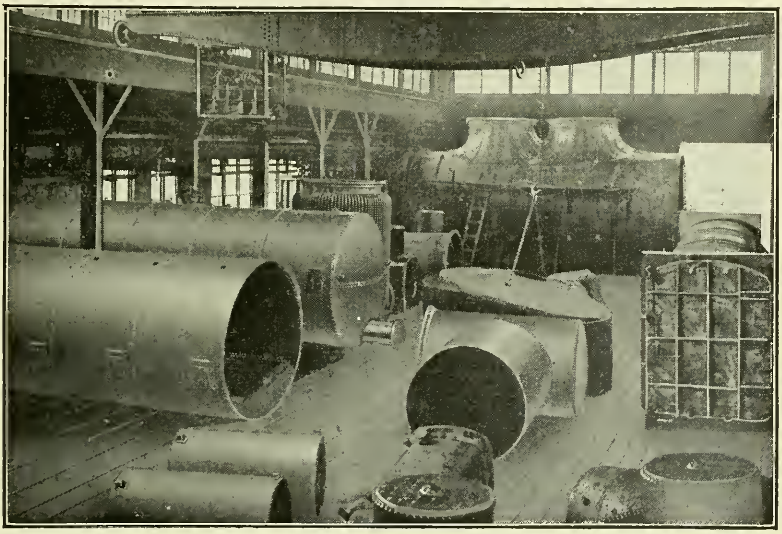
STRUTHERS-WELLS COMPANY

Established 1851

WARREN, PENNSYLVANIA

N. Y. Office, 50 Church St

ANYTHING IN STEEL PLATE CONSTRUCTION FOR BEET SUGAR REFINERIES PAPER MILLS, OIL REFINERIES, SOAP FACTORIES, WOOD ALCOHOL AND TURPENTINE PLANTS, TANKS (FOR STORAGE OR PRESSURE) WITH EITHER WELDED OR RIVETED SEAMS. PENSTOCKS, RIVETED PIPE FOR HYDRAULIC AND STEAM PRESSURE, EXHAUST PIPES, STEAM HEADERS, EXHAUST HEATERS, BRICK HARDENING CYLINDERS, GALVANIZING POTS, RETORTS, STILLS, AGITATORS, STEAM PANS, AND JACKETED KETTLES



View of one corner of Erecting Floor

We will contract to furnish specially designed work in steel plate from customers' plans and specifications, either completely built up in the shop or assembled, knocked down and shipped in sections as may be called for.

Our equipment and experience enable us to handle both intricate and simple work with equal ease and dispatch. Our workmanship will pass the most rigid inspection.

We maintain a fully equipped estimating department at our shops and also at our New York Office. This enables us to furnish prices promptly.

If you want a specially designed piece of work, send us your drawings and specifications,—we have probably built it before, or at least something very similar to it.

Smoke breechings, stacks and air flues for modern buildings are right in our line.

THE PETROLEUM IRON WORKS CO.

SHARON, PA.

NEW YORK ST. LOUIS HOUSTON, TEXAS

FABRICATORS AND ERECTORS OF EVERY VARIETY OF LIGHT AND HEAVY STEEL PLATE CONSTRUCTION.

TANKAGE FOR ALL PURPOSES LARGE OR SMALL; OIL REFINERY EQUIPMENT, STAND PIPES, WATER TOWERS, SMOKE STACKS, PENSTOCKS, RIVETED STEEL PIPE, BLAST FURNACES, HOT METAL LADLES, BOILERS, ANNEALING BOXES, "LEMAN" COUNTER CURRENT CONDENSERS, "GEM" FUEL OIL BURNERS, "WASHINGTON" AUTOMATIC OIL AND GAS SEPARATORS, PORTABLE RECEIVING TANKS, CAR TANKS ETC.

The Petroleum Iron Works Company is located on a property of forty acres, situated about three miles south of Sharon, Pa.

Our railroad facilities here are excellent, as the works have direct connections with the New York Central, the Erie and the Pennsylvania Railway Systems. Our trackage for receiving and shipping inbound and outbound freight is more than a mile in length, thus affording ample space for car storage and ideal facilities for making shipments. Car shortages and delays occurring where there is but one railroad connection are practically unknown in this district. We, therefore, feel sure the trade will fully appreciate this condition, as it enables us to guarantee our promises of delivery to points in every direction.

Our present plant consists of a substantial steel structure, fully equipped with modern machinery, tools and appliances, which insure accurate and rapid production. We are now in position to give our customers better satisfaction and more prompt service than ever before. Superior manufacturing facilities and careful shop inspection will continue to maintain our long established reputation for high class work.

It is difficult to prepare a catalogue that will adequately describe and illustrate all the classes of sheet metal structures and apparatus that we are in position to furnish and erect. The illustrations presented are shown merely as types of our various lines of work, among which we might enumerate the following:

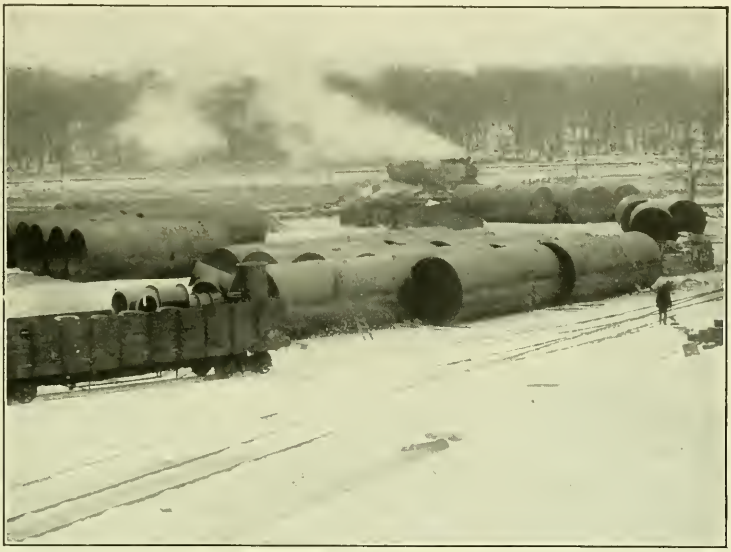
Oil Storage Tanks (large and small)
Acid Storage Tanks
Water Tanks
Water Softener Tanks
Molasses Tanks
Turpentine Storage Tanks
Grain Tanks
Tar Tanks
Hydro-Pneumatic Tanks
Pressure Tanks
Filter Tanks
Car Tanks
Portable Receiving Tanks
Railroad Water Service Tanks
Oil Refineries (complete)
"Leman" Counter Current Condensers
"Washington" Automatic Oil and Gas Separators
"Gem" Fuel Oil Burners
Riveted Steel Pipe
Flumes
Penstocks
Blast Furnaces (complete)
Converters
Hot Metal Ladles
O. H. Furnaces
Pulp Digesters
Galvanizing Kettles
Creosoting Cylinders
Condenser Boxes
Coal Bins
Steam Separators
Air Receivers
Water Towers
Stand Pipes
Smoke Stacks (Self-Supporting and Guyed)
Breechings, etc., etc.

Frequently special problems arise, and in such cases we are glad to offer the services of our Engineering Department. In order to assure prompt replies to inquiries we respectfully request complete detail information, including specifications and blue prints when possible. We hope by prompt and careful attention to our customers' inquiries to be favored with at least a portion of your valued orders.

We issue no discount sheet owing to the constant fluctuations in prices, and for the reason that practically all of our products are built to order.

THE PETROLEUM IRON WORKS CO.

OIL REFINERIES—COMPLETE



10′ x 30′ Single, Double, and Triple Compartment Cylindrical Horizontal Oil Storage Tanks ready for shipment.

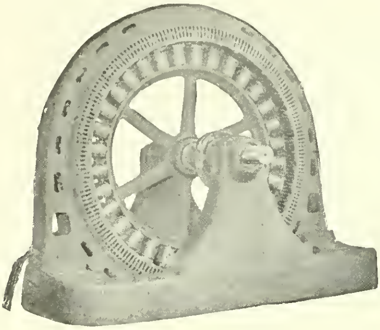
150 000 Gallon Water Tower, Yonkers, N. Y (Height over all 184′)

R. R. Water Service Tank. Erected East Waco, Texas, for the "Cotton Belt."

CROCKER-WHEELER COMPANY

AMPERE, N. J.

MANUFACTURERS AND ELECTRICAL ENGINEERS.
MOTORS, GENERATORS, TRANSFORMERS.



AC GENERATORS.

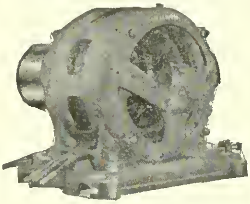
OF ALL SIZES.

Of superior design and construction. Special advantages gained by C-W exclusive features. Write for our bulletins 143, 135 and 150.

DC GENERATORS.

ALL SIZES.

These machines are known among engineers for the special care and attention given to every feature of their design and construction. Send for our bulletins 153 and 121.



INDUCTION MOTORS.

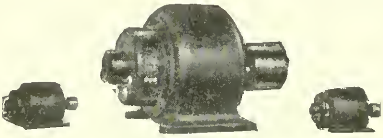
ALL SIZES.

These motors combine high efficiency, low starting current, great overload capacity and high power factor with strong insulation and ease of repairs. They will operate continuously with very little attention.

Bearings adjustable for wear.

Metallic slot wedges, allowing the use of form wound coils plus the advantages of closed slots.

Write for our bulletins 141 and 146.

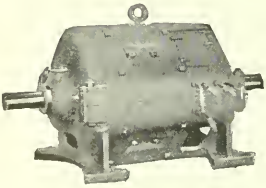


DC MOTORS OF ALL TYPES

SIZES FROM 120 TO 5000 H. P.

These motors are a C-W specialty and have been developed to stand hard service and abuse. Their quality is due to good design and thorough construction.

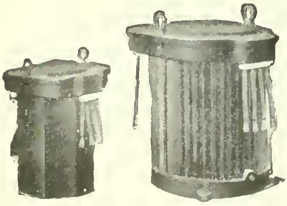
Send for our bulletin 142.



SPECIAL ROLLING MILL MOTORS.

Built with fire-proof insulation and all parts made extra strong and heavy for rolling mill service. The C-W Co. has made a specialty of this class of motors, and can promptly supply motors for any branch of Rolling Mill service.

Write for our bulletin 128.



POWER AND LIGHTING TRANSFORMERS.

These transformers are built for long service and efficiency. They operate with the minimum loss of energy, saving 15% or 20% of the amount lost by the average transformer.

Send for bulletins 151 and 154.

DIEHL MANUFACTURING COMPANY

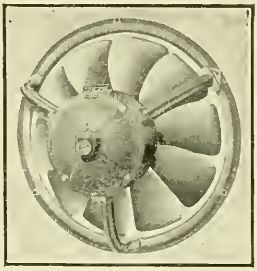
ELIZABETHPORT, N. J.

ELECTRICAL APPARATUS

VENTILATING OUTFITS

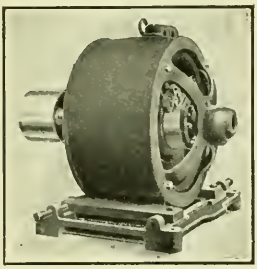
DIEHL MOTOR DIRECT CONNECTED TO EXHAUST WHEEL

Diehl
TRADE MARK



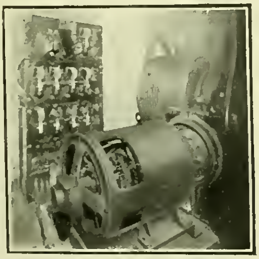
The illustration shows one of the many types which is carried in stock by the Diehl Manufacturing Company, who have standardized a slow speed totally enclosed motor for ventilating purposes. Particular emphasis is laid upon the enclosed type of motor from the fact that the entire volume of dust laden air handled by the wheel must pass over the motor and not through it.

TYPE "G" MOTORS



For general power uses this line of motors is exceptional for low temperature rise and sparkless operation within guaranteed load limits without change in brush position. Soft steel is used for the cast welded magnet cores. A vacuum impregnating process gives thorough saturation to the field coils with the insulating compound. All machining operations being performed on the yoke with one setting insures concentricity of all parts. Type "G" motors are made for Direct Current only and are of three varieties, open, semi-enclosed with wire screens and enclosed with iron doors.

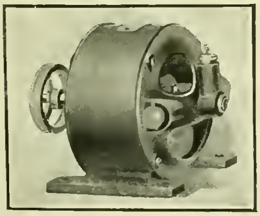
TYPE "K" MOTORS



Type "K" apparatus as built for adjustable speeds gives the full range of rated speeds by field resistance, having a culminating ratio of 5-1. This apparatus is more compact in its dimensions than any other on the market having similar characteristics, without sacrificing strength and durability.

Elevator motors subject to heavy starting torque, quick acceleration and sudden fluctuating load must blend the characteristics of wide speed range with sparkless commutation and low temperature rise. Type "K" applied to elevator drive gives positive assurance of success.

"GL" AND "GS" MOTORS



Small motors for general power uses showing a range of power from $\frac{1}{50}$ to $\frac{1}{4}$ HP are listed under the title of "GL" "GS" Motors in Bulletin No. 76.

The smaller sizes are constructed to run either D. C. or A. C. circuits. Having wholly laminated field frame and being enclosed with light sheet metal covers, weight is minimized. "GS" type motors are built for D. C. current only with either series, shunt or compound windings.

ELECTRO-DYNAMIC COMPANY

BAYONNE, NEW JERSEY

THE INTER-POLE MOTOR

The use of Direct Current Motors before the introduction of the INTER-POLE was rapidly decreasing, because the A. C. Induction Motor for Constant Speed Working had certain advantages, such as no attention being required to brushes or commutator.

The advent of the INTER-POLE Motor has changed this condition, because there is no commutator or brush trouble encountered with such—besides it has many other extremely useful characteristics not possessed by its predecessors for either A. C. or D. C. Circuits.

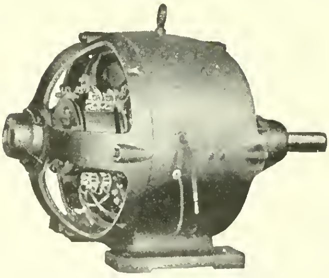
The INTER-POLE Motor, therefore, has fixed and established the basic principles on which all good D. C. Motors must be constructed. It constitutes the greatest advance in electric motive power in the last twenty years and HAS SAVED THE DAY FOR DIRECT CURRENT FOR MOTOR WORK.

This Company is the Pioneer of the INTER-POLE Motor development and has created a standard of merit by which all motors are judged.

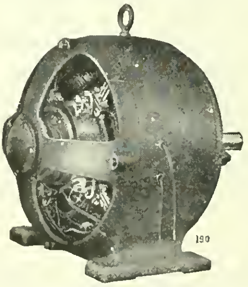
For either Constant or Adjustable Speed Service these motors have certain inherent advantages, among which are the following:

Type "S" vertical motor—Ball bearings.

1st: SPARKLESSNESS.
2nd: High Efficiency.
3rd: Reversibility.
4th: Heavy overload capacity.
5th: Compactness.
6th: Constant speed at any controller position regardless of load.
7th: Wide speed ranges. Speed ratio up to 1 to 6 by shunt field control.



Type "S" motor—commutator end
Ring oiler bearings



Type "S" motor—commutator end
Ball bearings.

A comparison of ratings and prices with other makes of motors will show that prices of the INTER-POLE Motors are competitive, even with motors not having INTER-POLES. This fact, together with the well-known high quality, will warrant the use of these motors exclusively.

Bulletins will tell the whole story. Send for them.

RICHMOND ELECTRIC COMPANY

RICHMOND, VIRGINIA

List of Branch Offices and Sales Representatives

Atlanta, Ga.	72 Marietta Street	Fulton Electric Co.
Baltimore, Md.	204 N. Calvert St.	John S. Dobler
Birmingham, Ala.	3 So. 20th St.	Oliver Electric & Machine Co.
Boston, Mass.	141 Milk Street	B. B. Stoddard
Chattanooga, Tenn.	Cor. Duncan Ave. & R.R. St.	Chattanooga Armature Works
Chicago, Ill.	322 Monadnock Block	Richmond Electric Company
Cincinnati, Ohio	126 W. 2nd Street	Buckeye Equipment Company
Cleveland, Ohio	505 Swetland Bldg.	The F. Bissell Company
Columbia, S. C.	804 W. Gervais Street	Gibbes Machinery Company
Dallas, Texas	1413 Commerce Street	Gus Sachs
Davenport, Iowa	119 E. 4th Street	Tri City Electric Supply Co.
Detroit, Mich.	1006 Majestic Bldg.	The F. Bissell Company
Louisville, Ky.	514 W. Main Street	Harry I. Wood Company
Lynchburg, Va.	614 Main Street	Hudson-Morgan Electric Co.
Memphis Tenn.	28 S. 2nd Street	Gray Brothers
Moline, Iowa		Tri City Electric Supply Co.
New Orleans, La.	Baronne & Peridodo Sts.	Interstate Electric Co.
New York, N. Y.	145 Chambers Street	Richmond Electric Company
Oklahoma City, Okla.	116 N. Broadway	Arnold & Wetherbee
Philadelphia, Pa.	1011 Chestnut Street	Richmond Electric Company
Pittsburg, Pa.	430 Penn Ave.	John McC. Price Company
Rochester, N. Y.	33 Exchange Street	L. Larsen
St. Joseph, Mo.	820 Frederick Ave.	Columbian Electric Company
St. Louis, Mo.	1318 Pine Street	Richmond Electric Company
San Francisco, Cal.	Monadnock Bldg.	A. C. Hanna
Toledo. Ohio	226 Huron Street	The F. Bissell Company

"RICHMOND" POLYPHASE MOTORS

By devoting their attention exclusively to the design of Polyphase Motors the engineers of the Richmond Electric Company have brought its product to an unusually high degree of perfection.

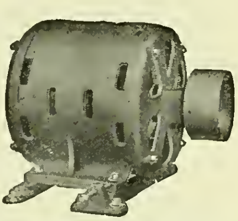
"Richmond" Motors are manufactured in both the Squirrel Cage and Wound Rotor types.

SQUIRREL CAGE MOTORS

The standard type R. E. (squirrel cage) motors are built in sizes from ½ to 150 h.p. Instead of auto-starters "RICHMOND" type R. E. motors are equipped with a special starting winding (covered by U. S. Patent of Feb. 12, 1907).

This form of starter has many advantages over the compensator method. "RICHMOND" Motors will develop a higher starting torque with less current than will motors using compensators.

Motors of 5 h.p. are supplied with these starting coils when so specified.



Illustrating 112 frame, 30 h. p. 1140 R. P. M. Squirrel Cage Motor.

WOUND ROTOR TYPES

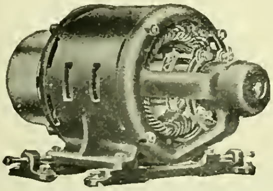
Type F. H. Slip Ring Motors

An appreciation of the demand for a satisfactory variable speed polyphase alternating current motor has caused this company to place upon the market a full line of machines of this character, ranging in size from one-fourth to 125 h.p. inclusive.

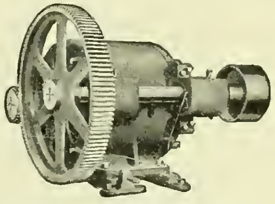
These motors will be found most satisfactory for the operation of printing presses, ventilating fans and other apparatus requiring variable speed.

These motors resemble very closely in operating characteristics a direct current shunt motor with armature control.

This type machine is most desirable for work requiring a powerful starting torque, as in this respect it is far superior to the squirrel cage motor.



50 h.p. 1140 r.p.m. 60 cycle wound rotor type motor. This gives a very accurate idea of the entire lines of type F.H. and E. M. slip ring motors in sizes from ¼ to 75 h.p.



Back Geared Motors can be supplied in all sizes in either the R.E. or F.H. types Standard ratio is 5-1.

Type E. M. Slip Ring Elevator Motors are rated for intermittent service. These motors are unusually well adapted for the operation of freight and passenger elevators, cranes, hoists, etc. In mechanical construction they resemble the constant service type F. H. motors, but have a higher starting torque.

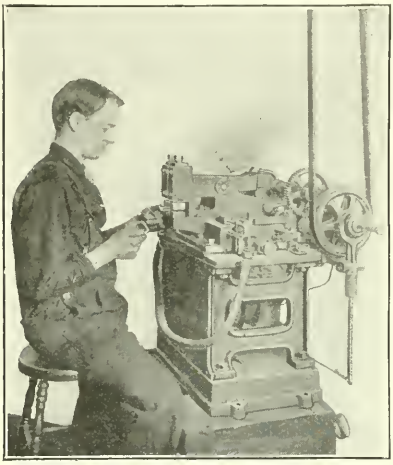
THOMSON ELECTRIC WELDING CO.

LYNN, MASS.

A COMPLETE LINE OF ELECTRIC WELDERS FOR BUTT, SPOT AND SPECIAL WELDING, OPERATED BY HAND, FOOT, SEMI-AUTOMATIC AND AUTOMATIC CONTROL.

SPECIAL WELDERS A SPECIALTY

TYPE 7A AUTOMATIC



For welding rings, hoops, buckles, etc. Machine-made in various forms, making 10-20 welds per minute.

COST, TIME, POWER

Iron-Steel Ro.	Area.	Horse-Power	K.W.	Seconds	Approx. K.W. hrs. 1000 Welds
¼	.05	5	4	2	2
⅜	.11	8	6	3	5
½	.20	12	9	6	15
⅝	.31	16	12	10	30
¾	.44	20	15	15	65
⅞	.60	24	18	18	90
1	.79	26	20	20	113
1⅛	.99	33	25	24	167
1¼	1.23	40	30	33	275
1½	1.77	50	38	40	422
1¾	2.41	64	48	48	640
2	3.14	80	60	60	1000

COST

Multiply the K.W. hours used by the number of cents your current costs per K.W. hour, and you get the current cost per 1000 welds for sizes shown on above table.

TIME

The time of welding runs from 1-5 second in small wire to 60 seconds in sections such as 3 square inches.

POWER

The power varies from 1 K.W. for 1-5 second to 60 K.W. for 60 seconds. Within certain limits the GREATER THE POWER, THE LESS THE TIME; and VICE VERSA. DIRECT CURRENT cannot be used, the welder being operated upon a single-phase of any constant potential ALTERNATING current, from 110 to 500 volts.

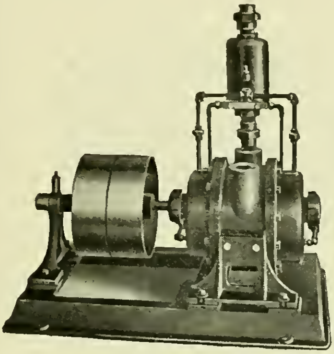
SPECIFICATIONS OF STANDARD TYPES

TYPE.	Approx. Weight	DESCRIPTION	DIMENSIONS Inches			Max. K.W. Trans. Capacity	Approx. Equivalent Horse-Power	WELDING CAPACITY				Approx. Time in Sec. for Welds	
	Lbs.		Length	Width	Height			Min. Inches		Max. Inches		Min.	Max.
								Ro	□''	Ro	□''		
1AA	90	Semi-Automatic	16	11	13	1.5	2	No. 23	wire	No. 15	.003		1
2A	150	Semi-Automatic	15	12	15	3.	4	No. 16	.003	¼	.05	1	2
2AA	140	Semi-Automatic	13	14	15	3.	4	No. 16	.003	No. 6	.03	1	2
2A	200	Automatic	16	24	17	3.	4	No. 16	.003	No. 6	.03	1	2
3A	500	Automatic	29	23	44	4.5	6	No. 10	.02	¼	.05	1	2
5A	525	Semi-Automatic	27	15	18	7.5	10	No. 8	.02	⅜	.11	1	4
5A	525	Hand	27	15	18	7.5	10	¼	05	⅝	.30	2	10
5AA	550	Semi-Automatic	27	15	20	7.5	10	No. 6	.03	¼	.05	1	2
7A	800	Automatic	31	24	36	6.	8	No. 8	.02	¼	.05	1	3
10A	900	Hand	32	16	32	15.	20	⅜	.11	¾	.44	4	15
10A	1000	T Welder	33	21	28	10.	13	$\frac{3}{16}$	.03	⅝	.30	2	10
20A	2500	Hand	48	30	36	30.	40	⅜	.11	1¼	1.23	4	33
20A	1500	T Welder	41	20	30	20.	27	¼	05	¾	.44	2	15
40A	7000	Hand	90	36	38	60.	80	¾	.44	2	3.14	15	60

CROWELL MFG. CO.

296-298 TAAFE PLACE BROOKLYN, N. Y.

SOLE MANUFACTURERS, UNDER PATENTS



Type D

CROWELL ROTARY AIR COMPRESSOR OR VACUUM PUMP

The action is *simple*, *positive*, and *durable*, with a continuous intake and delivery of the air. No valves, springs, gears, or unbalanced parts are used or required in the construction or operation of Crowell Machines.

The pressure or degree of vacuum is not dependent on high operating speeds. Floor space is small—no special foundations are required. Lubrication is positive and automatic.

Manufactured in nine standard sizes with capacities ranging from four to 390 cu. ft. of free air per minute, and designed for a working pressure of 25 lbs. or less to the sq. in.

When used as a vacuum pump, will maintain a vacuum of from 29 to 30 in.

Air Receivers supplied when required.

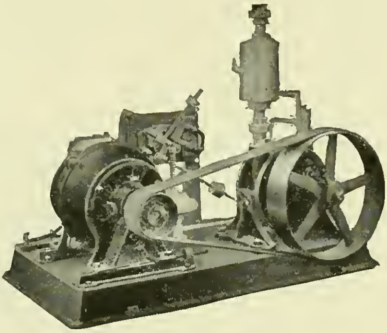
CROWELL ROTARY VACUUM PUMP

This machine was until recently used exclusively by the Paul System Co. in connection with their Patented vacuum system of steam heating, but is now available for all engineers and contractors.

It is electrically driven by a Sprague Round Type Motor and controlled by a rubber diaphragm switch, and can be set so that it starts and stops at any desired vacuum from 1 to 10 inches, thus automatically keeping the heating lines free from air and water.

Sizes from 1 to 5, having capacities sufficient for from 4000 to 60,000 sq. ft. of radiating surface.

Equally adapted for many other purposes requiring a higher vacuum.



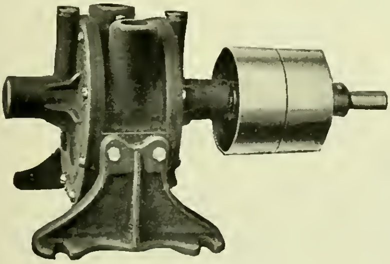
Type F

CROWELL POSITIVE PRESSURE BLOWER

These blowers are suitable for all purposes requiring air under pressure of from 1 to 10 lbs. per sq. in. or any degree of vacuum not exceeding 24 inches.

The principle of construction is identical with that of the machines described above; an internal drum and shaft carries sliding blades which operate in close contact with the cylinder wall. *No springs, valves, gears or unbalanced parts.*

Sizes of from 7 to 390 cu. ft. of free air per minute.



Type A. Sizes 1 and 2

Write for Catalogue.

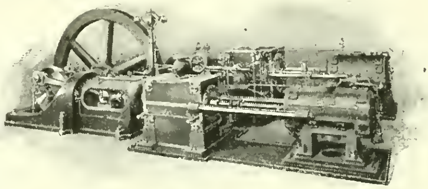
CHICAGO PNEUMATIC TOOL COMPANY

Fisher Building, CHICAGO 50 Church St., NEW YORK

MANUFACTURERS OF AIR AND GAS COMPRESSORS FOR ALL CLASSES OF WORK AND OF ANY CAPACITY OR METHOD OF DRIVE

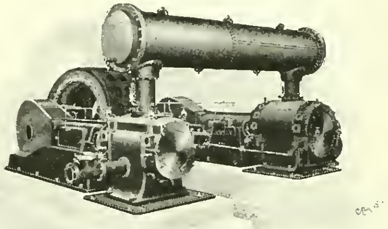
CLASS "C":

Heavy Duty Horizontal Cross Compound or Duplex Corliss, duplex or two-stage air cylinders, for steam pressures from 80 to 200 lbs., in capacities from 1500 to 5000 cubic feet.



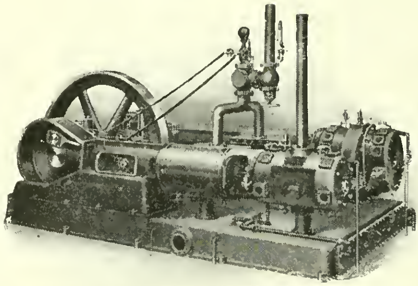
CLASS "M-CE":

Electric Motor Driven Compressor, with motor mounted directly on compressor shaft. For any current or voltage. Duplex or two-stage air cylinders, in capacities from 550 to 4000 cubic feet



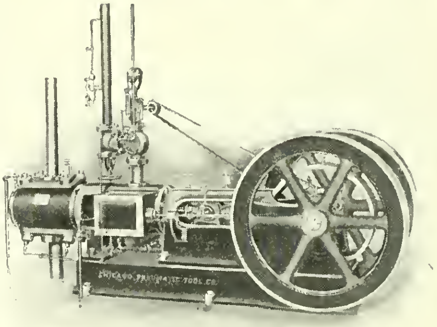
CLASS "G"-DUPLEX:

Duplex or Compound Steam Driven Type with duplex or two-stage air cylinders, with sub-base and intercooler in same. For steam pressures from 80 to 175 lbs., and in capacities from 190 to 2100 cubic feet.



CLASS "G"-STRAIGHT LINE:

Single Steam Single Air Cylinders in straight line, mounted on rigid frame, for steam pressures from 60 to 150 lbs., and capacities from 30 to 500 cubic feet.



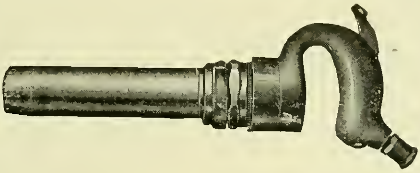
Also a full line of Belt and Short Belt Driven Compressors of corresponding types and sizes. Also gasoline engine driven compressors, stationary and portable, from 70 to 150 cubic feet capacity.

Descriptive pamphlets, specifications and prices on application to main offices or to branches in principal cities.

CHICAGO PNEUMATIC TOOL COMPANY

Fisher Building, CHICAGO
50 Church St., NEW YORK

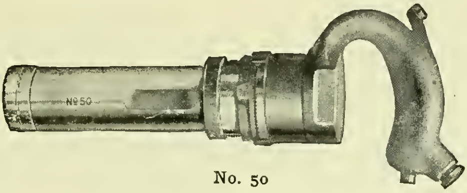
MANUFACTURERS OF PNEUMATIC HAMMERS FOR ALL PURPOSES, RIVETING, CHIPPING, CALKING AND CUTTING AND CARVING STONE



BK. CHIPPING AND CALKING HAMMERS

Made in seven sizes. Stroke lengths from one inch to four inches.

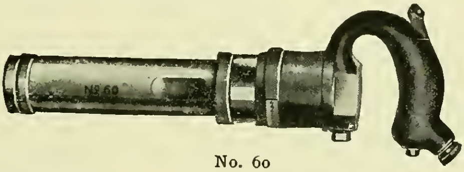
Catalogues and Bulletins on application.



No. 50

No. 50. RIVETING HAMMER

Weight 19½ lbs.; 5-inch stroke; capacity up to ¾-inch rivets.

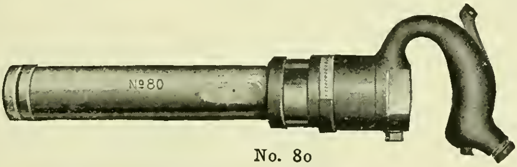


No. 60

No. 60. RIVETING HAMMER

Six-inch stroke; capacity up to ⅞-inch rivets.

Weight { Standard....22 lbs.
Heavy.....25 lbs.

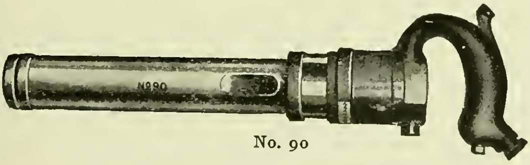


No. 80

No. 80. RIVETING HAMMER

8-inch stroke; capacity up to 1⅛-inch rivets.

Weight { Standard...24 lbs.
Heavy.....25 lbs.
Ex. Heavy..36 lbs.



No. 90

No. 90. RIVETING HAMMER

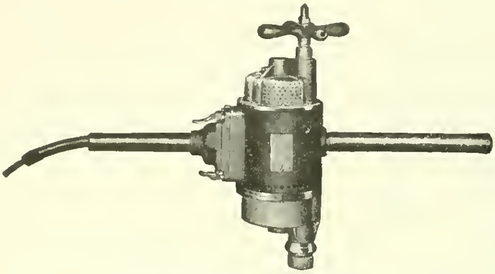
9-inch stroke; capacity up to 1¼-inch rivets.

Weight { Standard...25 lbs.
Heavy.....26 lbs.
Ex. Heavy..37 lbs.

CHICAGO PNEUMATIC TOOL COMPANY

Fisher Building, CHICAGO   50 Church St., NEW YORK

MANUFACTURERS OF PORTABLE ELECTRIC TOOLS, DRILLS, GRINDERS AND HOISTS FOR OPERATING ON DIRECT OR ALTERNATING CURRENT.



Size No. 2 S-S. Capacity $\frac{7}{8}$ inch in metal

UNIVERSAL ELECTRIC DRILLS

Operating on either direct or alternating current interchangeably.

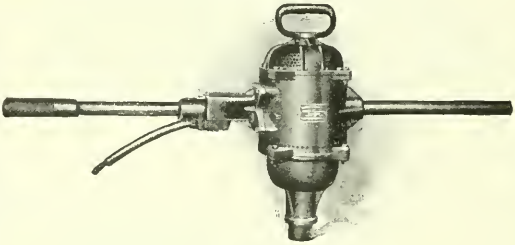
Size		Inch
No. 000.	Capacity in metal	$\frac{3}{16}$
No. 000x.	Capacity in metal	$\frac{1}{4}$
No. 00.	Capacity in metal	$\frac{5}{16}$
No. 0.	Capacity in metal	$\frac{3}{8}$
No. 1.	Capacity in metal	$\frac{1}{2}$
No. 2.	Capacity in metal	$\frac{7}{8}$
No. 3.	Capacity in metal	$1\frac{1}{4}$

HEAVY DUTY DIRECT CURRENT ELECTRIC DRILLS

For 110 and 220 volts.
Built in five sizes as follows:

Size		Inch
No. 0.	Capacity in metal	$\frac{3}{8}$
No. 1.	Capacity in metal	$\frac{1}{2}$
No. 2.	Capacity in metal	$\frac{7}{8}$
No. 3.	Capacity in metal	$1\frac{1}{4}$
No. 4.	Capacity in metal	2

The No. 3 and No. 4 sizes can be furnished in the center spindle as well as the side spindle style.



Style No. 4 C-S

HEAVY DUTY ALTERNATING CURRENT ELECTRIC DRILLS

For two and three phase.
Built in five sizes as follows:

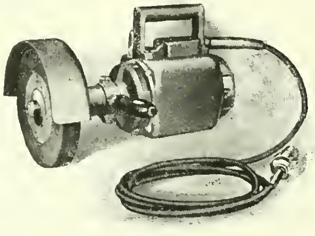
Size No. 0.	Capacity in metal	$\frac{3}{8}$ inch.
Size No. 1.	Capacity in metal	$\frac{1}{2}$ inch.
Size No. 2.	Capacity in metal	$\frac{7}{8}$ inch.
Size No. 3.	Capacity in metal	$1\frac{1}{4}$ inch.
Size No. 4.	Capacity in metal	2 inch.

Furnished in the side spindle style only. Standard windings are for 60 cycles, 110 or 220 volts. Sizes Nos. 2, 3 and 4 can be wound for 440 volts.

ELECTRIC GRINDERS

For use in the foundry, machine and structural shop. Built in two sizes for 110-220-600 volts direct current, and 110-220 volts two or three phase alternating current.

Size No. 5 BP carries 5x$\frac{3}{4}$ inch wheel.
Size No. 8 BP carries 6x$1\frac{1}{4}$ inch wheel.



Size No. 8 BP Portable Grinder

Bulletins on application.

CHICAGO PNEUMATIC TOOL COMPANY

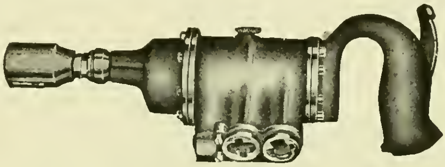
Fisher Building, CHICAGO — 50 Church St., NEW YORK

MANUFACTURERS OF IMPROVED BALL BEARING DRILLS OF ALL TYPES, FOR ALL CLASSES OF WORK WITHIN THE SCOPE OF A PORTABLE DRILL.

10F LITTLE GIANT MIDGET DRILL

The ideal drill for drilling tell-tale holes in stay-bolts and keeping them open.

Speed, light, 2400 R.P.M. Weight, 8 lbs.

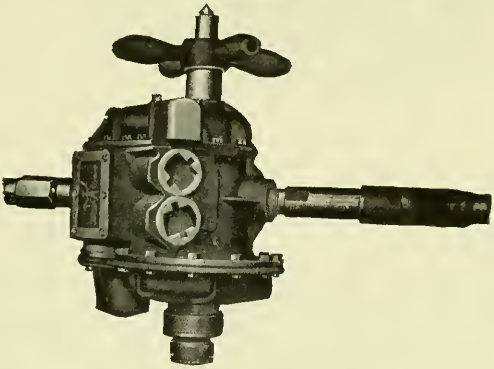


IMPROVED LITTLE GIANT BALL BEARING DRILLS

Size 1
Speed, 340 R.P.M.
Weight, 58 lbs.
Capacity, 4 Morse Taper.

Size 2
Speed, 450 R.P.M.
Weight, 41 lbs.
Capacity, 3 Morse Taper.

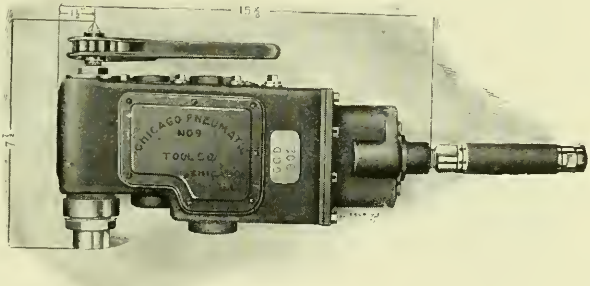
Size 4
Speed, 700 R.P.M.
Weight, 22½ lbs.
Capacity 2 Morse Taper.



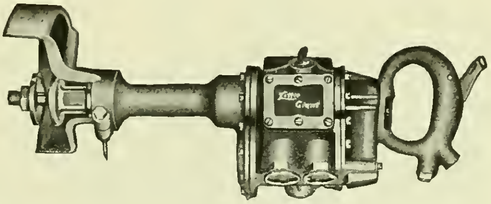
LITTLE GIANT CLOSE QUARTER DRILLS

Size 8
Speed, 270 R.P.M.
Weight, 38 lbs.
Capacity, 3 Morse Taper.

Size 9
Speed, 135 R.P.M.
Weight, 38 lbs.
Capacity, 4 Morse Taper.



No. 4 IMPROVED LITTLE GIANT GRINDER



As will be seen from the cut, the grip handle is provided with a trigger which when pressed down opens the throttle, and is held in that position until released by pressing a button at the side.

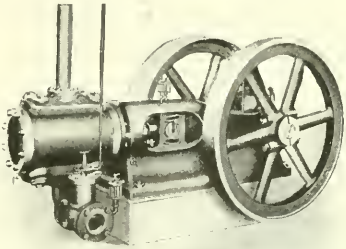
Speed 3000 R.P.M. Capacity, 8″ x 1″ Emery wheel.

Catalogues and Bulletins on request

THOS. H. DALLETT CO.

York and 23d Sts., PHILADELPHIA, PA.

MANUFACTURERS OF AIR COMPRESSORS, PNEUMATIC STONE-WORKING TOOLS, BOILER SCALERS, WOOD CARVING TOOLS, ETC.



Straight Line—Belt Driven

"Dallett" Compressors are made in straight-line, duplex and compound types, belt, steam and motor driven, their characteristics adapting them for any service where the requirements call for a combination of economy, high efficiency, ease of operation and maintenance.

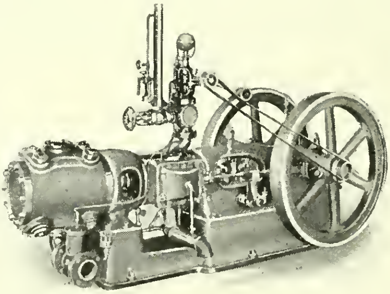
The standard machines range in capacities from 79 cu. ft. per minute up; all types being compact and massive in construction, with large bearing surfaces, producing a machine of strength, reliability and efficiency.

Frames are of the open fork, center-crank type, with a large amount of metal under the main bearings, and all parts requiring adjustment are easily accessible.

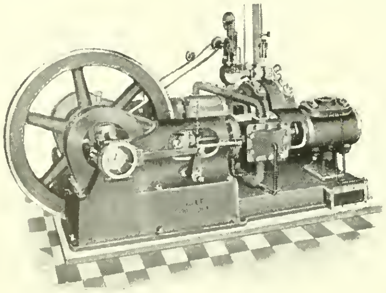
All air and steam cylinders are adapted for pressures up to 125 lbs., the steam valves being of the plain D slide valve type, but on the larger machines the Meyer Adjustable Balanced Cut-off Valve is used where specified.

All air cylinders are equipped with the "Dallett" Silent Inlet and Discharge Valves, these being of the Poppet type, allowing a full intake and discharge of air to the capacity of the cylinder at each stroke of the piston.

The steam passages are direct; air passages and valve ports are of ample area, combined with large water-cooling surfaces and thorough and efficient lubrication, and losses by clearance and leakage have been reduced to minimum.



Straight Line—Steam Driven



Duplex—Steam Driven—Two-Stage Air

All machines are equipped with automatic controlling and unloading devices. Compound machines with two-stage air cylinders are furnished with inter-cooler of large cooling surface, employing the return flow type of water circulation.

All ratings have been conservatively made, and every machine is given an exhaustive test above rated load before leaving the shop.

WRITE FOR BULLETINS.

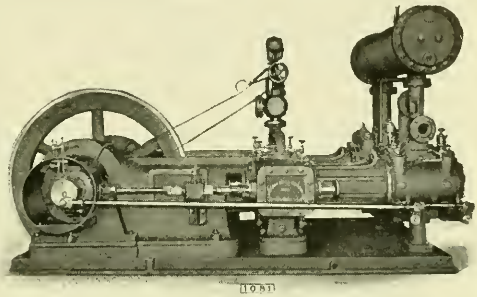
LAIDLAW-DUNN-GORDON COMPANY

115 BROADWAY, NEW YORK

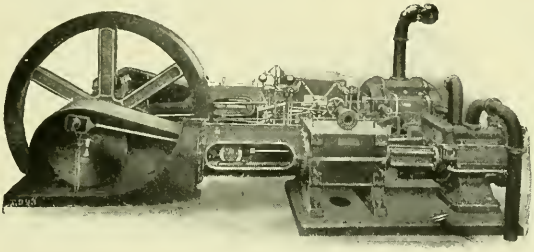
WORKS: Cincinnati, Ohio. Branch Offices in all Principal Cities

AIR COMPRESSORS, VACUUM PUMPS, HIGH DUTY PUMPING ENGINES

AIR COMPRESSORS. 300 to 10,000 cu. ft. per minute. 10 to 5000-lbs. air pressure. Our highly developed production system both in Foundry and Machine Shop, together with careful standardization and interchangeability of parts, permits our giving to our clients the highest obtainable value for their money. Our Cincinnati Air Valve Gear as continuously developed and improved by us, has set the pace in the air compressor business for this country for ten years, and permits our giving to our clients the highest obtainable efficiency. Both value and efficiency, as well as the nature of our service to our clients, are attested by the highest proportion of repeat orders received by any builder in the compressor business.



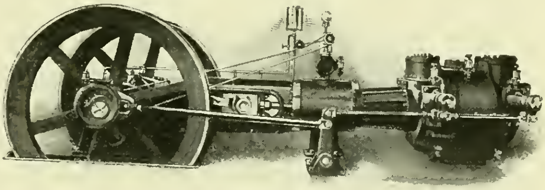
Meyer Gear Air Compressor



Corliss Air Compressor

Bulletin L521-68A describes our compressor line in condensed form.

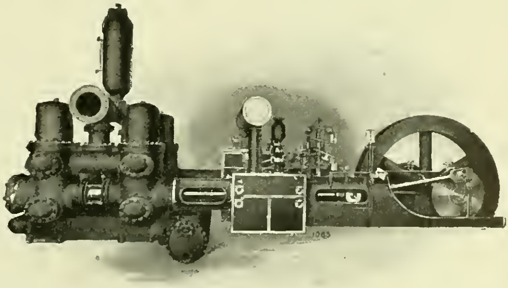
VACUUM PUMPS. Our close clearance high efficiency rotary dry vacuum pump serves four out of five of the largest turbine plants in the country. Our gear is so designed that continued use improves efficiency. We have no flash ports.



Vacuum Pump

HIGH DUTY PUMPING ENGINES. One million to twelve million gallons capacity. We are replacing direct acting plants of the older type, in large numbers, with high duty installations, operating on one-third to one-half of the previous fuel expenditure.

Our business policy is based on the effort to so completely satisfy a client, as to insure our receiving his next order.



High Duty Pumping Engine

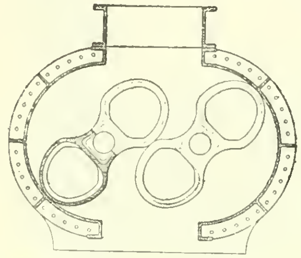
L 172.8

P. H. & F. M. ROOTS COMPANY

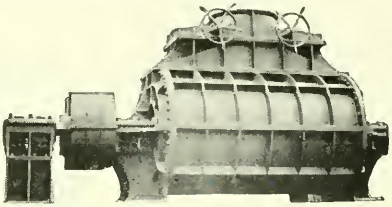
CONNERSVILLE, INDIANA

ROTARY BLOWERS, GAS EXHAUSTERS, ROTARY PUMPS FLEXIBLE ROPE COUPLINGS

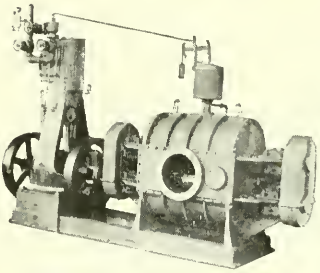
ROOTS' BLOWERS



Sectional View Showing Interior Construction



Double Outboard Bearing, Single Geared Smelting Blower with Double Acting Quick Opening Blast Gate



Engine Driven Gas Exhauster

FACILITIES

SIZE.—Our facilities are such that we can build Blowers up to sixty-inch gear machines, having a displacement of over seven hundred cubic feet per minute.

ACCURACY.—By the use of machines specially made for our work, we insure accuracy, which results in high efficiency, not once, but in all machines.

CAPACITY.—Is the largest of any Positive Blower Company in the world.

SPECIAL MACHINES

We build special machines for handling gases. They can be bronze or lead lined, with or without stuffing boxes. The general construction is the same as Smelting or High Pressure Blowers.

GAS EXHAUSTERS

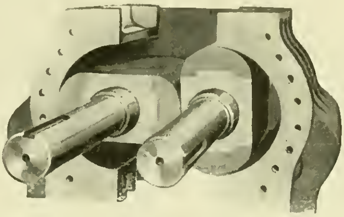
The Roots' Exhauster as built today, is the nearest approach to a perfect Rotary machine. It is strong throughout, without being cumbersome, work manlike in appearance, without sacrificing strength and durability to finish, easy to adjust, and has all adjustments in places easy of access, without removing one part to get at another.

TABLE OF SIZES, POWERS, AND CAPACITIES OF ROOTS' GAS EXHAUSTERS

No. of Exhauster	Suction and Discharge Diameters	Horse Power at Stated Speed and One Pound Pressure	Speed of Exhauster	Displacement in Cubic Feet per Revolution	Capacity per Hour in Cubic Feet No Allowance for Shrinkage
2	4	.75	200	.91	10,920
3	6	1.5	200	1.31	15,720
4	8	2.5	180	2.95	31,860
5	10	3.75	170	4.8	48,960
6	12	5.	160	8.3	79,680
7	16	7.50	150	13.1	117,900
8	16	11.	140	19.6	164,640
8½	20	15.5	130	28.2	219,960
9	20	19.	120	38.5	277,200
9½	20	24.	110	51.2	337,920
10	24	29.	100	61.75	370,500
10½	30	36.	95	81.	461,700
11	30	50.	90	111.2	600,480
11½	36	69.	90	147.9	798,660
12	36	80.	85	192.	979,200
12½	42	102.	85	244.	1,244,400
14	42	115.	80	304.9	1,463,520

P. H. & F. M. ROOTS COMPANY

ROOTS ROTARY PUMPS



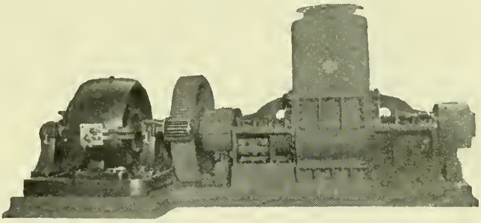
Interior Construction of Rotary Pump

The range of operation of these Pumps is from ten feet to two hundred feet head and for handling any liquid substance, not containing grit, with economy ranging from 75 per cent to 85 per cent of the power applied to the Pump shaft.

The source of power may be turbine, steam or motor, and any of these classes of power may be direct connected, geared or belt driven, thus giving a wide choice of arrangement and allowing the Pump to be used under almost any conditions.

Irrigation reservoir, condenser, cooling towers, circulating work for hot or cold water or brine, are a few among the many purposes to which the Pumps are successfully applied. The sizes range from the 50,000 gallons a minute irrigation pump, running 70 r.p.m., tc the small 1-10 gallon displacement per revolution Pump, running 600 r.p.m.

For handling oil, tar or ammonia, these pumps are well suited. Special lining and impellers may be used where the liquids to be pumped would attack cast iron.



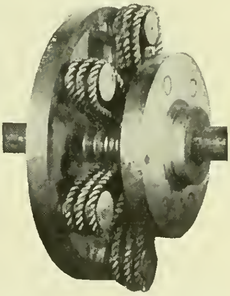
Direct Connected Motor and Pump

Briefly, the operation of the Pumps is as follows: The revolution of the shafts and impellers traps the water between the lobes and the case, delivers it to the discharge side, where the rolling together of the impellers on the center lines of the shaft prevents the return of the water.

The care and workmanship with which the impellers are fitted to each other and to the case assure a small slip of water both at the rolling and case contacts. The suction air chambers cast on the lower half of the half circles save unsightly pipe connections, at the same time stiffening the Pump as a whole.

The flow of water is steady and uniform, and unless air enters the suction pipe, free from pulsation.

FLEXIBLE ROPE COUPLINGS



This coupling was designed to meet a condition in rotary pump work, and proved so satisfactory for this work that it is now used on a large percentage of our direct connected Blowers, Exhausters and Pumps. The points of advantage that place it ahead of other designs for the same purpose are these:

FIRST—It corrects for misalignment in any direction, whether due to settling of foundations, wear of engine bearings, or original setting, thus saving uneven wear and heating.

SECOND—It takes care of end thrust caused by heating, wear of bearings or oscillations of the driving mechanism.

THIRD—It eases up sudden fluctuations of load by the swing of the loop.

FOURTH—It permits rotation in either direction, with equal results and symmetrical positions.

FIFTH—The ropes can be removed as quickly as the bolts in a solid coupling can be taken out.

SIXTH—With ropes off, either the driving or driven machine can be rotated without interference.

SEVENTH—The life of the ropes is long and can be renewed at small expense.

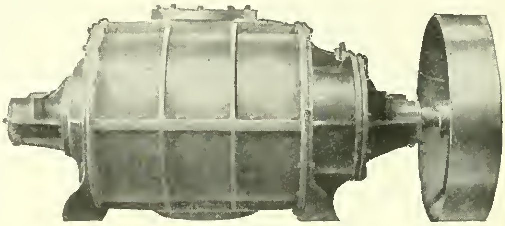
WILBRAHAM-GREEN BLOWER CO.

POTTSTOWN, PA.

MANUFACTURERS OF THE "WILBRAHAM-GREEN" ROTARY POSITIVE PRESSURE BLOWER AND GAS EXHAUSTER. KNOWN EVERYWHERE AS "THE OLD RELIABLE." SUITABLE FOR EITHER LOW OR HIGH PRESSURES. HIGH PRESSURE MACHINES ARE, HOWEVER, USUALLY DESIGNED TO SUIT THE PARTICULAR WORK THEY ARE TO PERFORM.

A few of the lines of work for which our Blowers and Gas Exhausters are especially adapted and have been used during the last 40 years: Smelters, Pneumatic Tubes, Oil Burning, Foundry Cupolas, Paper Mills, Vacuum Pumps, Oil Refineries and Gas Works.

The internal construction of Blowers and Gas Exhausters is very similar.

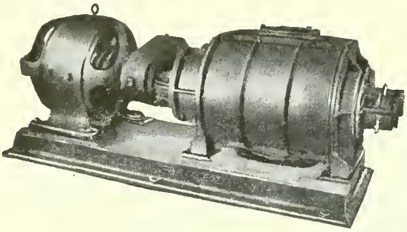


Pulley Driven Blower

Low pressure machines are usually single-geared.

High pressure machines are usually double-geared.

Bearings are bushed with Phosphor Bronze; are Ring Oiling, and very large, so that the pressure per inch square of projected area is extremely low.



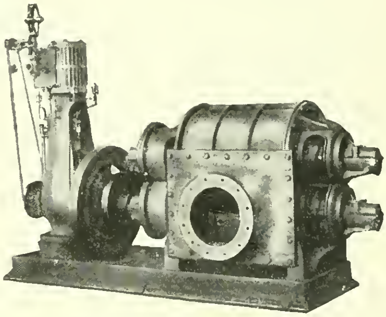
Motor Driven Blower

Shafts are FORGED STEEL.

Our design allows the driving pulley or gear to be placed against the end of the Bearing, reducing the overhang to a minimum. This is a very IMPORTANT feature, in either a Pulley or Motor Driven unit.

Gears are accurately cut from the solid and very wide face.

Blowers or Exhausters are built with pipe connections either top or bottom or on the sides, to suit conditions.



Engine Driven Exhauster

Regular Pulley-Driven Blowers. Approximate Dimensions in Inches

No.	Displacement per Revolution Cubic Feet	Total Length	Vertical Blower Side Outlet		Horizontal Blower Top or Bottom Outlet			PULLEY		Average Shipping Weight
			Total Height	Face to Face of Flanges	Total Height	Face to Face of Flanges	Total Width	Diameter	Face	
1	3	50	31	23	24	23	30	20	5	1,525
2	5½	67	31	23	24	23	30	24	6	2,100
3	8	70	36	26	27	26	36	30	6	2,800
4	13	78	42	29	30	29	40	32	7	3,800
5	19	87	47	34	34	34	46	40	8	4,800
5A	22	99	48	34	36	31	47	40	9	5,700
5B	25	107	48	34	36	34	47	42	9	6,100
6	29	100	56	41	40	41	55	44	10	7,800
6A	35	110	58	42	43	42	57	48	10½	9,500
6B	45	120	63	44	45	41	62	48	12	11,400
7A	55	130	70	48	48	48	67	51	12	15,500
7B	67	124	80	60	61	60	81	60	15	17,500
7½	85	144	80	60	61	60	81	66	15	19,000
8	112	154	90	64	66	64	89	72	18	32,000

Larger Sizes Built to Order

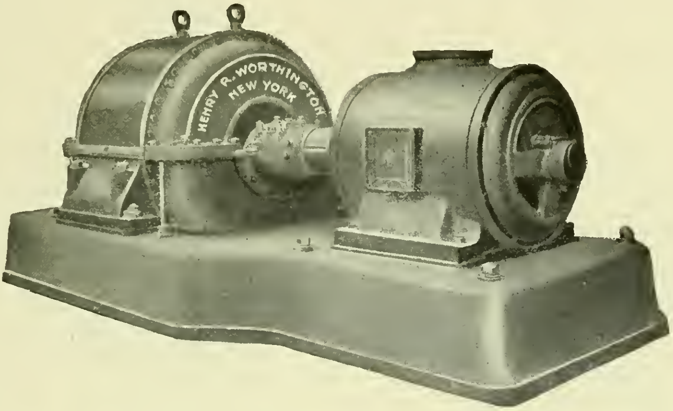
HENRY R. WORTHINGTON

115 Broadway, NEW YORK　　Works: HARRISON, N. J.

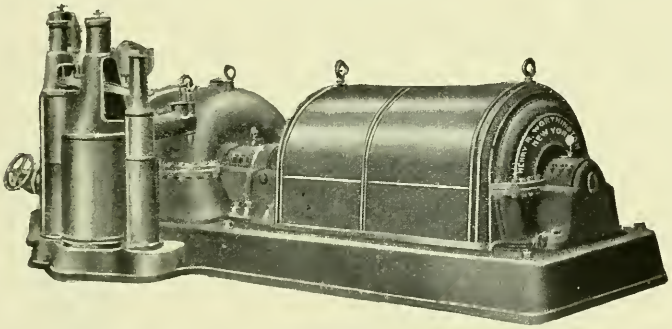
Manufacturers of

Turbine Driven and Motor Driven TURBO-BLOWERS and TURBO-COMPRESSORS

For every pressure and capacity



Motor Driven Turbo-Blower—Capacity 30,000 cu. ft. per min. 15 lbs. gage pressure



Turbo-Compressor driven by Mixed Pressure Turbine
Capacity 10,000 cu. ft. per min. 100 lbs. gage pressure

After careful investigation the Worthington Company have acquired the exclusive rights for the manufacture and sale on the American continent and in the Hawaiian and Philippine Islands, of the full line of Turbo Compressors and Blowers so successfully developed by Messrs. Pokorny & Wittekind, of Frankfort, a.m., Germany,—and are in position to build and supply Turbo Compressors and Blowers for all ranges in capacity and pressure ordinarily covered by reciprocating machines.

Purchasers may be assured that this line of Compressors has absolutely passed beyond the experimental stage. It has produced the highest economy ever attained and has thoroughly demonstrated that it is the most practical and reliable apparatus of its type now in the market.

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W 206.8

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BATTLE CREEK, MICH.

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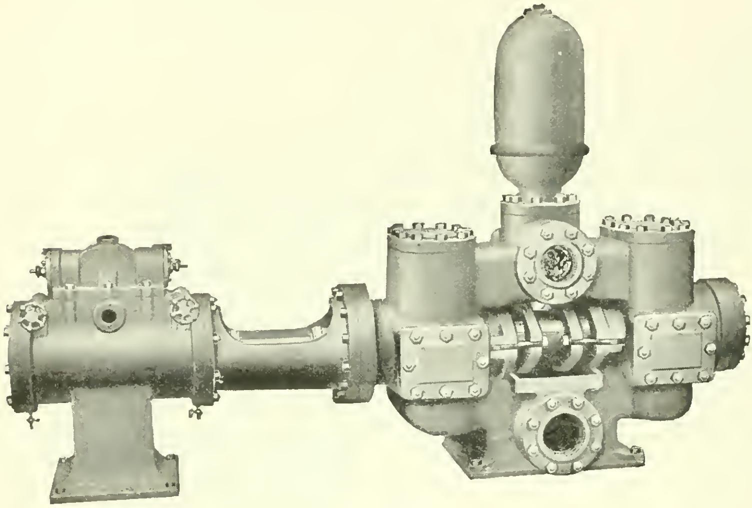
MARSH STEAM PUMPS AND AIR COMPRESSORS, AMERICAN SIMPLE AND COMPOUND PUMPS, AMERICAN POWER PUMPS AND AIR COMPRESSORS.



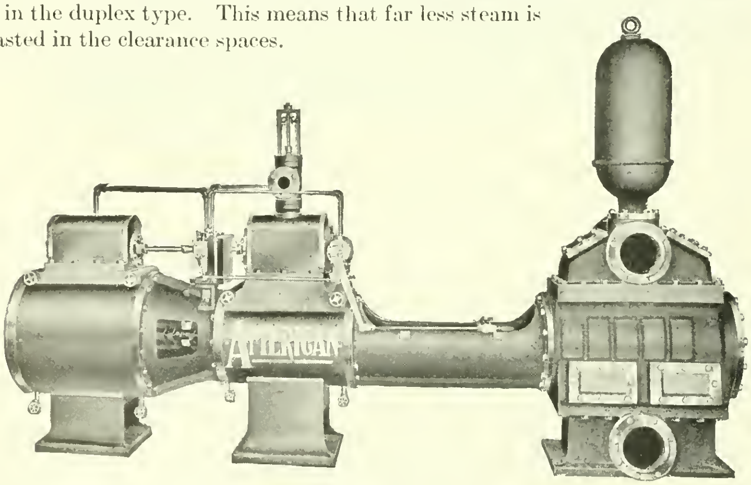
Marsh and American Steam Pumps are of the "Simplex Type" and offer the following advantages over the "Duplex" type built by other manufacturers:

(a) Having fewer parts they are less liable to break-down.

(b) Making only half as many strokes and stops they waste less steam.

(c) Having only two steam ports as compared with eight in a duplex pump less steam is wasted in the ports.

(d) They have only two clearance spaces as compared with four in the duplex pump, and the full length of stroke is always assured, instead of being variable as in the duplex type. This means that far less steam is wasted in the clearance spaces.



AMERICAN STEAM PUMP COMPANY

BATTLE CREEK, MICH.

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- Air Lift Pumping
- Air Pumps (vac.)
- Automatic Feed Pumps and Receivers
- Automobile Pumps
- Beer Pumps
- Boiler Feed Pumps
- Brine Pumps
- Combined Air and Circulating Pumps
- Combined Air and Feed Pumps
- Condensers, Jet
- Condensers, Surface
- Creamery Pumps
- Deep Well Pumping Engines
- Directions
- Drip Pans
- Dry Vacuum Pumps
- Filter Press Pumps
- Fire Pumps
- General Service Pumps
- High Service Pumps
- Hydraulic Pressure Pumps
- Hydraulic Pumps
- Magma Pumps
- Marine Pumps
- Milk Pumps
- Model Pump
- Naphtha Pumps
- Oil Pump, Special
- Plunger Pumps, Outside Packed
- Power Milk Pumps
- Pumping Outfits
- Pump Stands
- Receiver Pumps
- Sinking Pumps
- Sugar House Pumps
- Sweet Water Pumps
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- Vacuum Pan Pumps
- Water Cylinders, Deep Well
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  - Heavy Service, Belted
  - Light Service, Belted
  - Motor Driven
- Air Compressors, Steam
- Compound Pumps
  - Piston, Heavy Service
  - Piston, Light Service
  - Plunger, Outside Center Packed
- Power Pumps
  - Heavy Service, Belted
  - Heavy Service, Geared
  - Light Service, Belted
  - Light Service, Geared
  - Motor Driven
- Simple Pumps
  - Piston, Heavy Service, Short Stroke
  - Piston, Heavy Service, Long Stroke
  - Piston, Heavy Service, Special
  - Piston, Light Service, Short Stroke
  - Piston, Light Service, Long Stroke
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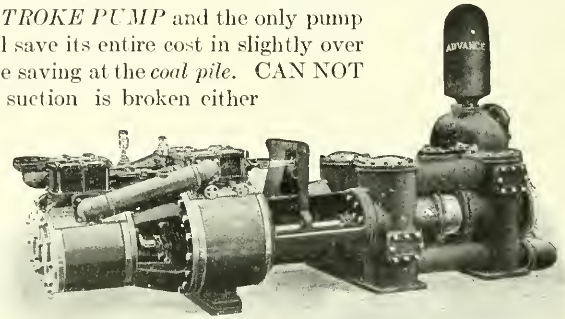
SPECIAL CHARACTERISTICS. EVERY DUPLEX PUMP *IS A FULL STROKE PUMP* and the only pump of this type which will save its entire cost in slightly over one year's time, by the saving at the *coal pile*. CAN NOT RUN AWAY in case suction is broken either with or without load.

OUR VALVE MOTION is of special construction, insuring LONG LIFE to this part of the pump which for years has been the chief trouble with duplex pumps.



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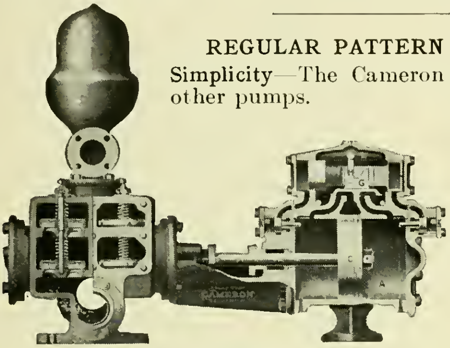
REGULAR PATTERN FOR GENERAL SERVICE.

Simplicity—The Cameron has fewer working parts than other pumps.



Sectional View of Cameron Regular Pattern Piston Pump.

Durability — The steam mechanism consists of four stout pieces only. None of these parts are delicate, intricate or exposed to hazard or possible injury.

When under full steam pressure the suction is lowered or lost, the Cameron can be operated at full speed with less damage or danger of the piston striking the cylinder heads than any other pumps on the market.

No Outside Valve Gear—The steam valve movement works in line with the piston rod without the intervention of arms or levers. There are no rods to become bent, broken or get out of alignment, no tappet bars, rollers or clamps to adjust. Therefore, the Cameron can be run at a higher speed than other pumps with less danger, yet increased capacity.

Stroke—A Cameron Pump will not reverse until it has completed its full stroke.

Wear—Every Cameron Pump is packed to compensate for wear.

Maintenance and Repairs—The Cameron will give the maximum of capacity and service with the minimum cost of maintenance and repairs.

DIMENSIONS AND CAPACITY OF CAMERON REGULAR PATTERN PISTON PUMPS FOR GENERAL SERVICE

Size Number	Diameter of Steam Cylinder, Inches	Diameter of Water Cylinder, Inches	Stroke of Piston Inches	Capacity per Stroke Gallons	Capacity at Ordinary Speed per Minute Gallons	Steam Pipe	Exhaust Pipe	Suction Pipe	Discharge Pipe	Floor Space Inches	Weight
0	3½	2	4	.054	8	⅜	½	1¼	1	32 x 9	136
1	4	2	6	.081	12	⅜	½	1¼	1	40 x 10	210
2	5	2½	6	.12	18	½	¾	1½	1¼	40 x 11	260
3	6	3	7	.21	28	¾	1	2	1½	47 x 13	418
3a	6	3½	7	.29	38	¾	1	2½	2	47 x 15	435
4	7	3½	7	.29	38	¾	1	2½	2	47 x 15	459
4a	7	4	7	.39	50	¾	1	2½	2	51 x 16	457
5	7	3½	12	.5	50	1	1½	3	2½	58 x 17	820
5b	7	5	13	1.10	100	1	1½	4	3	63 x 20	1117
6	8	4	12	.65	65	1	1½	3	2½	58 x 18	864
6a	8	5	13	1.10	100	1	1½	4	3	63 x 20	1160
7	10	5	13	1.10	100	1¼	2	4	3	64 x 21	1345
8	10	6	13	1.59	150	1¼	2	4	3½	64 x 21	1411
9	12	7	13	2.16	200	1½	2½	5	4	66 x 24	1928
10a	14	8	13	2.83	261	2	3	5	5	73 x 26	2548
10	14	9	18	4.96	330	2	3	6	5	81 x 30	3126
11	16	10½	18	6.75	450	2½	4	8	6	90 x 37	4920
12	18	12	20	9.80	587	3	4	10	8	103 x 41	6080

THE DEMING COMPANY

SALEM, OHIO, U. S. A.

NEW YORK OFFICE AND STOCK: 152 CHAMBERS STREET

SINGLE AND DOUBLE ACTING TRIPLEX PUMPS; ARTESIAN WELL PUMPS, FOR OPERATION BY ELECTRIC MOTORS, GAS OR GASOLINE ENGINES, OR BELT FROM SHAFT.

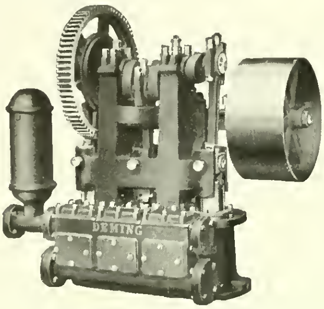


Fig. 50, Size 7 x 8 to 8½ x 8.

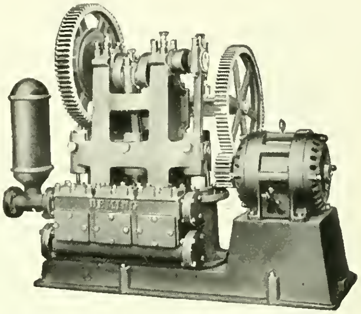


Fig. 50, Size 5½ x 8 with Type "B" Drive.

FIG. 50, STANDARD SIZES, CAPACITIES, ETC.

PLUNGERS		CAPACITY			Maximum Working Pressure Pounds	DIAM. OF PIPES		Gear Ratio	Tight and Loose Pulleys
Diam. Inches	Strok In.	Gallons per Rev.	Usual Revs. per Min.	Gallons per Min.		Suction Inches	Dischg. Inches		
2	2	.081	70	5.67	150	1½	1	5 to 1	8 x 2
2½	2	.127	70	8.89	150	1½	1	5 to 1	10 x 2
2½	3	.19	60	11.4	150	2	1½	5 to 1	12 x 3
3	3	.27	60	16.2	150	2	1½	5 to 1	14 x 3
3½	3	.37	60	22.	150	2	1½	5 to 1	16 x 3
3½	4	.50	60	30.	150	2½	2	5 to 1	16 x 4
4	4	.65	60	39.	150	2½	2	5 to 1	18 x 4
4	6	.98	60	59.	160	2½	2	5 to 1	20 x 5
4½	6	1.24	60	74.	150	3	2½	5 to 1	20 x 5
5	6	1.53	60	9[illegible].	150	3	2½	5 to 1	24 x 5
5½	8	2.46	60	147.	150	4	3	5 to 1	28 x 6
6	8	2.94	55	161.	140	4	3	5 to 1	30 x 6
7	8	4.00	55	220.	150	5	4	5 to 1	30 x 8
8	8	5.22	55	287.	150	5	4	5 to 1	36 x 8
8½	8	5.90	55	324.	140	6	5	5 to 1	36 x 8
9	10	8.26	50	413.	160	8	6	5 to 1	42 x 10
10	10	10.20	45	459.	150	8	6	5 to 1	42 x 12
12	12	17.62	42	740.	150	10	8	5 to 1	48 x 16
13	14	24.10	40	964.	140	12	10	5 to 1	48 x 20

FIG. 62, CAPACITIES.

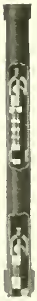


Fig. 324 Artesian Well Cylinder

Diam. and Stroke of Cylinder	CAPACITY		†Max. Depth of Well, Feet
	Usual Revs. per Min.	Gallons per Min.	
2¾ x 10	40	10.2	300
2¾ x 16	35	14.3	300
2¾ x 24	28	17.2	350
3¾ x 10	40	19.1	175
3¾ x 16	35	26.7	175
3¾ x 24	28	32.1	190
4¼ x 10	40	24.5	130
4¾ x 16	35	42.9	100
4¾ x 24	28	51.5	120
5¾ x 16	35	62.9	70
5¾ x 24	28	75.4	80
6¾ x 24	28	104.0	60
7¾ x 24	28	137.2	45

† Refers to a vertical distance from surface of water to point of delivery.

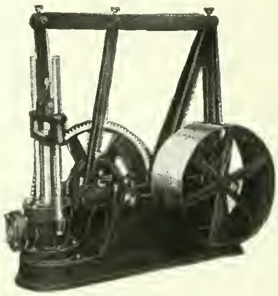


Fig. 62, 10-inch Stroke.

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EFFICIENT TRIPLEX AND CENTRIFUGAL POWER PUMPS FOR EVERY SERVICE

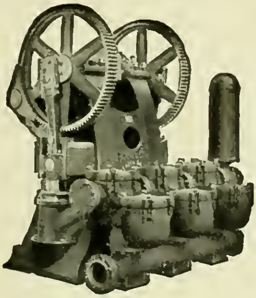

Figure 1283

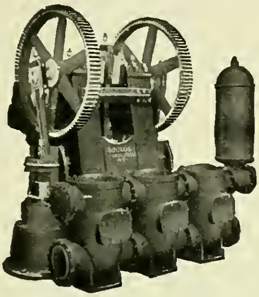

Figure 920

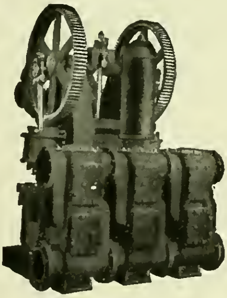

Figure 1364

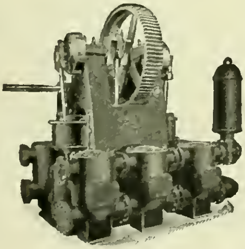

Figure 1590

Boiler Feeding
Elevator Service
Circulating
Semi Fluid
Drainage
Irrigation
Air Pressure
Fire Service
Chemicals
General Water Supply
Mines
Pulp Grinders
Railway Water Stations
Tank Pumping
Bleacheries
Dye Houses
Tanneries
Refineries
Oil Pipe Lines
Hydraulic Presses
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Sprinkler Systems
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Cyanide Pumping
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Ask for
Bulletins on Power or
Hand Pumps for any
service in which you
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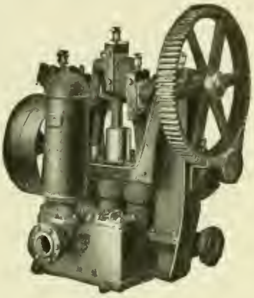


Figure 924

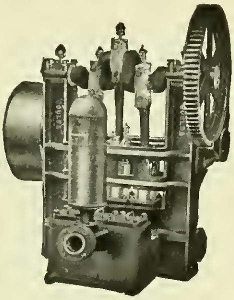


Figure 1009

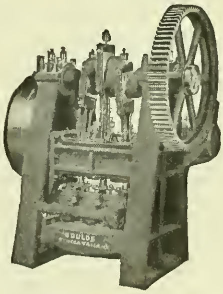


Figure 997—4½ x 8, No. D

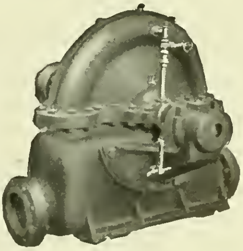


Figure 3001, No. 6

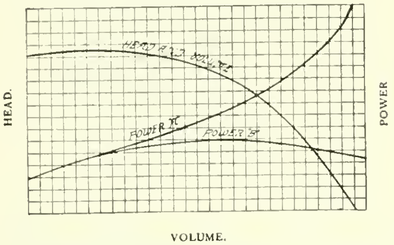
MANISTEE IRON WORKS CO.

MANISTEE, MICH.

Sole manufacturers and licensees for the United States and Canada for the

REES ROTURBO PATENT PRESSURE CHAMBER PUMP, ROTARY JET CONDENSER, AND AIR PUMP.

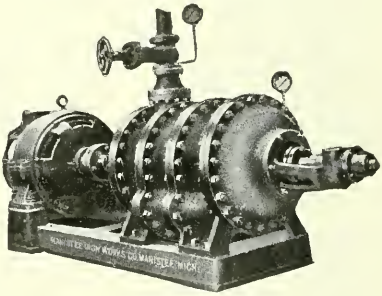
COMPARISON OF POWER CURVES.
A.—ORDINARY TURBINE CENTRIFUGAL PUMP.
B.—REES "ROTURBO" CENTRIFUGAL PUMP.

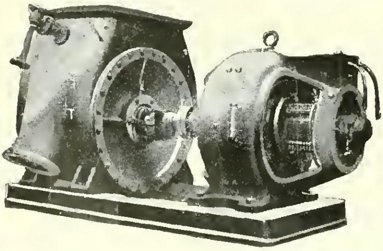


The distinctive features of these pumps are shown by the curves given. The H.P. taken by the Rees Roturbo Pump is considerably less than that taken by the ordinary turbine Centrifugal pump. This is shown clearly by the difference between the curves "A" and "B." The result of this is that the efficiency curve of the Rees Roturbo is sustained at a high figure over a wide range.

We are building these pumps for any service and type, either belt-driven, steam engine, electric motor or turbine; also for pumping to any height and delivering any quantity of water that may be required.

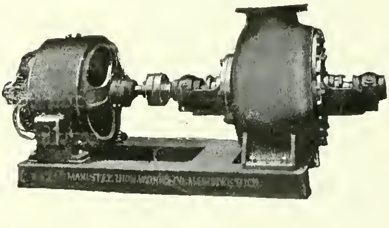
The illustration at the side is of a Rees Roturbo High Lift Colliery Pump delivering 60,000 gallons per hour. 570 ft. 1140 r.p.m. 79% efficiency.





The illustration here shows a Rees Roturbo Low Lift Pump delivering 30,000 gallons per hour. 28 ft. head, 900 r.p.m. 78% efficiency.

The figure here shows a Rees Roturbo Rotary Jet Condenser. These can be driven by either Electrical Motors, Steam Engines or Steam Turbine.



The vacuum attained is within a fraction of an inch of the theoretical possible. The condenser is extremely simple, having one Rotating Shaft and impeller only. There are no other moving parts whatever. The Rees Roturbo Rotary Air Pump is on the same principle. Full details and quotations on application.

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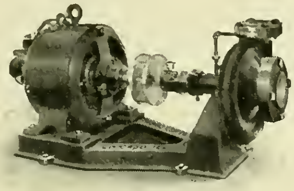
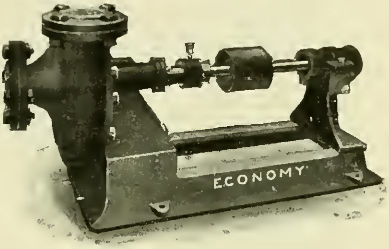
Manufacturers of

"ECONOMY" PUMPING EQUIPMENT

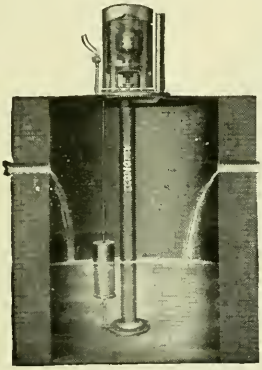


Motor Driven Centrifugal Pump



Belt Driven Centrifugal Pump

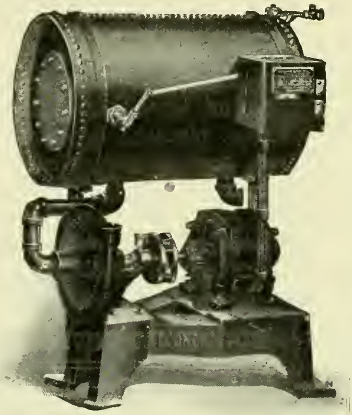
"Economy" Centrifugals are made in single and multi stage for any capacity at any head, both belted and direct connected to motor or engine. They are of the highest grade, having machined parts and castings, and are being used efficiently for pumping liquids containing large percentages of solids.



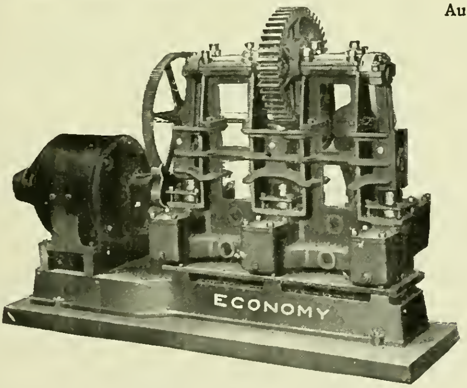
Bilge Pump or Sewer Ejector

The "Economy" Automatic Bilge Pump or Sewage Ejector is made for any capacity at any Head. Either single or duplex arrangement for the Submergible or Dry Basin Type.

They are also used for other purposes such as pumping various liquids from retorts, cisterns and tanks.



Automatic Condensation Pump and Receiver



"Economy" Triplex Pump

Note the compact, yet sturdy design. Built for heavy duty.

The "Economy" Automatic Condensation Pump and Receiver is connected with the return line, near or at any distance from the boiler and is especially essential where there are low places in the pipe lines or low radiators that collect the condensation.

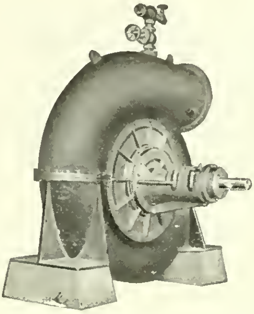
There is an "Economy" **Pump** to meet every demand. Forty-three different designs.

Write for complete catalog and a set of specifications covering each type.

MORRIS MACHINE WORKS

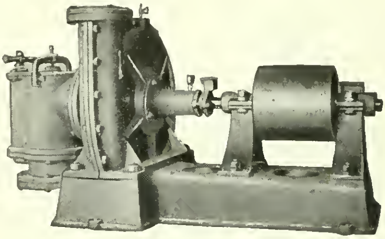
BALDWINSVILLE, N. Y.

CENTRIFUGAL PUMPING MACHINERY, STATIONARY AND MARINE ENGINES AND HYDRAULIC DREDGES.



Morris Centrifugal Pumps are simple in construction: they have only rotary parts, perfectly balanced, no reciprocating parts or valves; require small space and foundation; have high efficiency; are equally suitable for small up to very large capacities, and can handle sand or solids with the water without injury. These pumps direct connected to reciprocating engines are suitable for moderate heads, or direct connected to electric motor or steam turbine (or belt driven) for high heads. For heads above about 100 feet, pumps are built in stages.

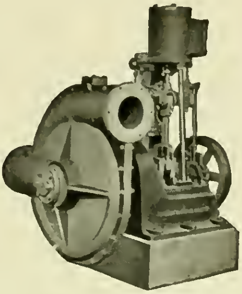
When making inquiries for pumps, full information should be given—that is, quantity desired, head, including friction (or give actual elevation and length of suction and discharge piping), type of pump desired, how driven—whether belt, steam engine, electric motor (give electric current characteristics), arrangement of suction and discharge openings desired, whether right hand or left hand, etc.

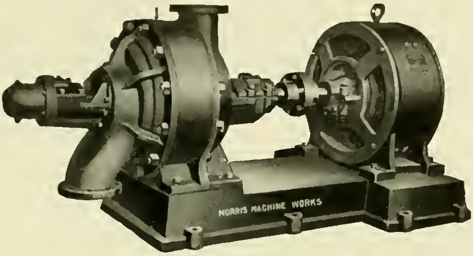


MORRIS IMPROVED STANDARD IRON HORIZONTAL PUMP.

No. Pump (Diameter Discharge Opening)	Size Pipe Flange on Sucton, Inches	Economical Capacity, for each Gallons per Minute	Horse-Power Required Pulley, Foot Head	Diameter and Face of pulley in Inches	Floor Space Required in Inches, Without Primer	Shipping Weight Without Primer, Lbs.	Shipping Weight With Primer, Lbs.	No. Pump
1½	2	70	.058	6 x 6	17 x 31	175	220	1½
1¾	2	90	.075	7 x 8	21 x 32	260	305	1¾
2	3	120	.10	8 x 8	23 x 37	350	415	2
2½	3	180	.15	8 x 8	24 x 38	360	430	2½
3	4	260	.22	8 x 8	25 x 39	415	495	3
4	5	470	.30	10 x 10	29 x 41	615	720	4
5	6	735	.45	12 x 12	34 x 54	940	1075	5
6	8	1050	.59	15 x 12	37 x 55	1180	1345	6
8	10	2000	1.00	20 x 12	45 x 64	2065	2430	8
10	12	3000	1.52	24 x 12	51 x 69	2610	2940	10
12	15	4200	2.00	30 x 14	63 x 71	3615	....	12
15	18	7000	3.50	40 x 15	77 x 80	7100	....	15
15	18	7000	3.50	30 x 15	60 x 68	3150	....	15
18	20	10000	4.50	40 x 16	93 x 103	9000	....	18
18	20	10000	4.50	30 x 16	66 x 72	4835	....	18
20	22	12000	5.40	36 x 20	73 x 83	6800	....	20
22	24	13000	5.50	48 x 20	126 x 130	....	....	22
24	24	15000	6.50	48 x 36	94 x 137	....	....	24

MORRIS MACHINE WORKS

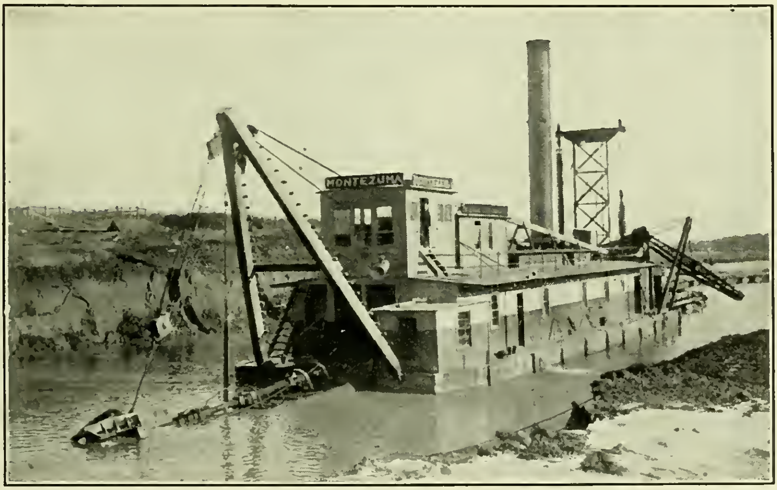




We build CENTRIFUGAL PUMPS for almost any service and of all types, including side suction and double suction, vertical or horizontal shaft. STAGE PUMPS for high heads. TWIN PUMPS for large capacities and high speeds. Or will design SPECIAL PUMPS to suit special conditions. We also build a very complete line of STATIONARY AND MARINE ENGINES, in single cylinder, compound and triple expansion types.

DREDGING PUMPS.

MORRIS Dredging Pumps are made in sizes from 2″ discharge and upward, both lined and unlined. They are belt driven or direct connected to steam engines. We can furnish pumps only or the complete dredge including all machinery.



20″ HYDRAULIC DREDGE with 750 H.P. MORRIS Triple Expansion Engine, Water Tube Boilers, Cutter Machinery. This Size Dredge Has an Average Capacity of 250,000 Cubic Yards of Material per Month.

R. D. WOOD & COMPANY

PHILADELPHIA, PA.

ENGINEER, IRON FOUNDERS AND MACHINISTS; PUMPING ENGINES CENTRIFUGAL PUMPS, HYDRAULIC MACHINERY, GAS HOLDERS AND APPLIANCES, GENERAL MACHINERY, CAST IRON PIPE, HYDRANTS AND VALVES, GAS PRODUCERS, GAS WASHERS

PUMPING ENGINES

Vertical Triple Expansion, and Direct Acting for Water Works, Sewage, Irrigation and for high pressures. High duty pumping engines of both the crank and fly wheel and direct-acting types. Designed to combine highest economic duty and efficiency with greatest reliability and utmost simplicity.

Estimates and drawings (either exact or preliminary) furnished upon application, with statement of requirements to be fulfilled.

CENTRIFUGAL PUMPS

Of most modern design and highest efficiency, for Water Works, High Pressure Systems, Irrigation and general service.

Following is the classification of our Standard Pumps:

SERVICE

Class C.	Compound Multi-stage Pumps for High Heads.
Class H.	High Lift Pumps, Single Stage, 40 feet and upwards.
Class M.	Medium Lift Pumps, Single Stage, 20 to 40 ft.
Class L.	Low Lift Pumps, Single Stage, 0 to 20 feet.
Class D.	Dredging Pumps, Single Stage, 0 to 60 feet.
Class T.	Pulp Pumps, Single Stage, 0 to 40 feet.
Class Q.	Special Pumps, Single Stage, in connection with above.

CAST IRON PIPE

Bell and Spigot Pipe from 1 inch to 84 inches in diameter, Flange Special deep bell, High Pressure, Flexible joint for submarine work, and every form of pipe and special casting, as well as heavy Loam Castings.

HYDRAULIC MACHINERY

Hydraulic Presses of every description for the heaviest work, Punches, Shears, Riveters, Intensifiers, Hoists, Pressure Pumps, Cranes, Valves, etc. etc. For the majority of operations to which hydraulic power can be applied, and especially those requiring very great force exerted through a comparatively short stroke, as in riveting, punching, shearing, lifting, forging and flanging, there is no other system at all comparable with it for efficiency, uniformity, simplicity or economy. This is true for several reasons; primarily in that there is absolutely no motion or power consumed except in the act and at the moment of performing the desired operation.

R. D. WOOD & COMPANY

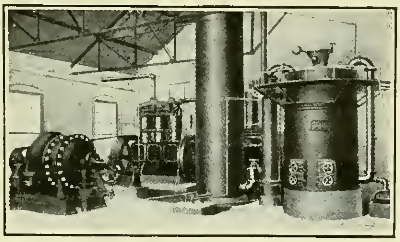
HYDRANTS AND VALVES

Fire Hydrants, Mathews patents for standard and high pressure. Gate, Check, Foot and Air Valves, Valve Boxes, Indicator Posts, Foot Valve and Intake screens, Hook Racks, etc.

PRODUCER GAS PLANTS

We have had years of experience in the building of producers for all kinds of fuel purposes as well as for power, and our customers may be certain of securing apparatus suitable to their requirements both from an economic and operating standpoint.

Our engineering department is at your service, and we would be pleased to have our representative visit your plant and give full details.



GAS HOLDERS AND APPLIANCES

Gas holders of largest dimensions and all machinery and appliances relating to gas works.

SINGLE OR MULTIPLE LIFT GAS HOLDERS

With or without steel tanks.

MITCHELL SCRUBBER

Matton & Mitchell Self-Sealing Mouthpieces, Purifiers, Condensers, Scrubbers, Bench Work, Center Seals, Gas Valves, Lamp Posts, etc.

GAS WASHERS

We control for the United States the Theisen Gas Washing Process, which we build for producer and blast furnace gas. This Process was adopted by the United States Steel Company at Gary, and is being put in with all their new construction. It delivers the gas to an engine cleaner than the air in the mixture.

GENERAL MACHINERY

Our shops are well equipped for building large machinery of every description, such as sugar, chemical and similar work.

INQUIRIES

For Pipe should state size, kind, approximate quantity and weight or pressure under which they will be used; and if possible the intended service, also delivery desired.

WILSON-SNYDER MFG. CO.

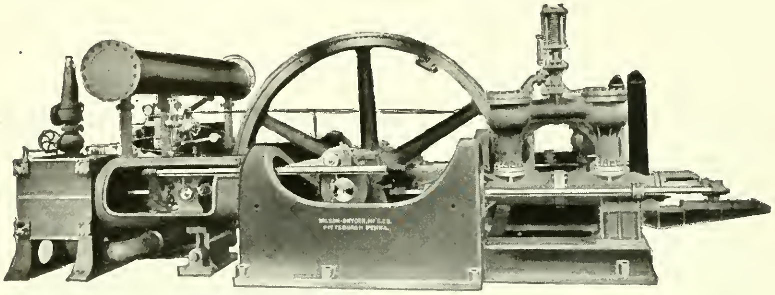
General Office and Works

PITTSBURGH, PA.

STEAM PUMPS
Rugged

CENTRIFUGAL PUMPS
Readily Accessible

ELECTRIC PUMPS
Accurate Workmanship



PUMPING ENGINE FOR PIPE LINE SERVICE
Normal capacity 20,000 barrels per day against 1000 lb. pressure.

Our standards are based on long stroke as the cardinal principle. So far as possible the interchangeable system of construction has been adopted. Bronze bushings and burrs, for example, are all interchangeable. We need hardly dwell on the immense advantage which this system offers. Instead of waiting, perhaps, weeks for the making of a trivial repair part, you have merely to telegraph or write to us for a new piece.

PARTIAL LIST OF WILSON-SNYDER PRODUCTS

- Accumulators
- Air Compressors
  - Direct Acting, High Pressure
- Air Pumps
  - Direct Acting, Independent with Jet Condenser
- Automatic Steam Regulating Valve
- Automatic Water Regulating Valve, for Pump Pit
- Boiler Feed Pumps, all types
- Centrifugal Pumps
  - Large volumes, low heads
  - Medium volumes, high heads.
- Compound Duplex Pumps
  - Piston, Plunger, Condensing, Non-Condensing, Hydraulic, Vertical and Horizontal
- Direct Acting Pumps
  - Air, Boiler Feed, Mine, Piston, Plunger, Tank or Light Service
- Duplex Power Pumps, all services
- Duplex Pumps, for all services
- High Duty Pumping Engines
  - Horizontal Cross Compound
  - Vertical Cross Compound
  - Vertical Triple Expansion
- Hydraulic Compound Duplex Pumps
- Hydraulic Crank and Flywheel Pumps
- Independent Air Pumps and Jet Condensers
  - Direct Acting, Single
- Jet Condensers
- Mine Pumps
  - Direct Acting, Duplex, Centrifugal
- Piston Pumps
- Direct Acting, Duplex
- Plunger Pumps
  - Compound and Duplex, Triple, Direct Acting
- Power Pumps
  - Duplex, Triplex, Vertical and Horizontal
- Regulating Balanced Valve for Accumulators
- Regulating Valve for High Pressure Pumps
- Steam Separators
- Steam Valves for Direct Acting Pumps
- Tank or Light Service Pumps
  - Direct Acting, Duplex
- Tar Pumps
- Triple Expansion Pumps
- Triplex Power Pumps for Hydraulic Service
- Vacuum Pumps
- Vertical Compound Duplex Pumps
- Vertical Cross Compound Pumps
- Vertical Triple Expansion Pumps

THE HILL PUMP VALVE CO.

CHICAGO, ILL.

HILL PUMP VALVES FOR BOILER FEED, VACUUM, ELEVATOR AND MINE PUMPS, PUMPING ENGINES, TAR, OIL AND BRINE PUMPS

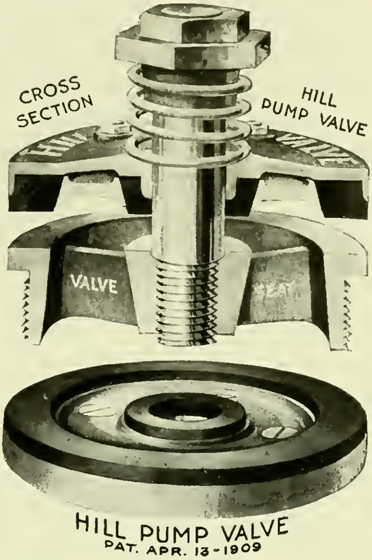
The only valve that *stays* efficient.

Hill Valves always seal tight, wear evenly and cannot ride on the bridges.

Pumps equipped with this type of valve give highest possible efficiency and capacity and continue to do so.

Installation of Hill Valves increases a pump's efficiency from 10 to 50%.

Made in all sizes 2″ to 8″ diameter.



CONSTRUCTION

The construction of this valve is shown in the cut. The valve consists of a bronze body carrying two composition seal rings, the seal rings being retained in place by the outer and inner lips of the body and by a concentric wedge ring. The body has a central sleeve bearing which fits loosely on the stem. The inner ring does not come in contact with the stem. The binder ring is bronze and is held to the valve body by brass screws and lock nuts. All similar parts of valves of the same size and type are interchangeable.

The valve seat has three ribs except in the larger sizes. The ribs of the seat are not required to support the valve and thus the lesser number increase the area of the waterway, and greatly decrease the friction. The bridges are machined below the seating surface so that the water stands above them even when the valve is on its seat.

The valve stems are Tobin bronze. The upper end is threaded for the spring cap and lock nut. The spring cap is grooved to center the spring.

The spring is made of a superior grade of phosphor bronze wire of suitable tension to seat the valve properly.

The lift of the valve is such as to give a free waterway area equal to or slightly in excess of the free waterway area of the seat.

Many pumps have seats which conform sufficiently to the Hill Valve to obviate the necessity of changing seats.

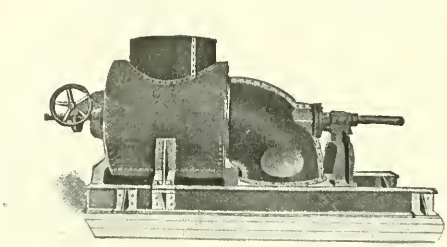
For further information address the Hill Pump Valve Co., Chicago.

HOLYOKE MACHINE COMPANY

HOLYOKE, MASS.

WATER WHEELS AND GEARING

SHAFTING AND MILL WORK, PUMPS, HYDRAULIC PRESSES, PAPER AND WOOD PULP MACHINERY, ETC.



Horizontal Turbine

Vertical Turbine

WATER POWER DEVELOPMENTS

The Holyoke Machine Co. has complete engineering and manufacturing facilities for the development of water power plants of all kinds. We are prepared to furnish designs and machinery of standard or special pattern, and are always glad to bid for work in our line. The services of our engineering staff are always at the service of prospective purchasers.

We manufacture Boyden and Hercules Turbine Wheels, both vertical and horizontal, with flumes and all connections. Water Wheel Governors and Head Gates to suit all conditions and all necessary gears, shafts, pulleys, and Mill Work.

BEVEL, MITER AND SPUR GEARS

Our gears are made of Charcoal Iron of such mixture as insures maximum strength and durability, making them far superior to Gears made from ordinary Anthracite Iron. Our collection of gear patterns is one of the largest in the country, and all patterns are kept in first-class condition. We have pattern-makers constantly employed in making new gear patterns, and can usually furnish gears from special patterns to order in a short time.

PLANED GEARS. For many years we have made a specialty of Planed Gears, and with large facilities for such manufacture, we are prepared to furnish both Bevel and Spur, Angle and Mortise Gears, finished with exactness and promptness. Catalog on request.

OTHER PRODUCTS

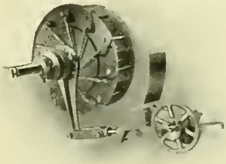
Other manufactures of the Holyoke Machine Company include: Improved Paper Machinery and Machinery for Wood Pulp Mills, Cotton, Paper, and Husk Rolls. Chilled Iron Rolls, Machinery for Bleacheries and Dye Works, Rotary Fire Pumps, and Friction Drives, Power Pumps, Hydraulic Presses and Pump.

Special Machinery and Iron Castings of every description made to order.

RODNEY HUNT MACHINE COMPANY

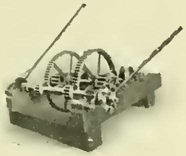
ORANGE, MASS., U. S. A.

TURBINE WATER WHEELS, FLUMES, PENSTOCKS, CANAL AND RESERVOIR GATE APPARATUS, ROTARY, TURBINE AND CENTRIFUGAL PUMPS, HEAVY POWER TRANSMISSION EQUIPMENTS, TEXTILE WET FINISHING MACHINERY



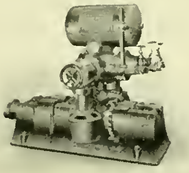
HUNT IMPROVED DRAW BAR GATE APPARATUS

This illustration shows the adaptability of improved apparatus in connection with our Hunt-McCormick Turbines when mounted on a horizontal shaft. The sliding draw bar, extending thru the packing bearing with brass bushing located in bracket casting riveted to outside of cylindrical flume, is provided with a machine cut cogged rack on outer end. A small spur gear operates the draw bar rack and a larger spur gear on the same shaft has a pin clutch on the side. The main gate shaft, extending over the clutch spur gear, operates same with small pinion spur gear so that engaging or disengaging operation of the gearing when running is readily and quickly effected by use of a pin clutch.



HOISTING AND HEAD GATE APPARATUS

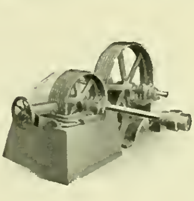
Having one of the largest assortments of both Hoisting Apparatus and Head Gate Apparatus, we are able to offer designs meeting practically every requirement both for Gates and Hoisting Mechanisms. Also Wood Bulk Head Gates, etc. See Catalog 29, Section H-G.



ROTARY FIRE PUMPS

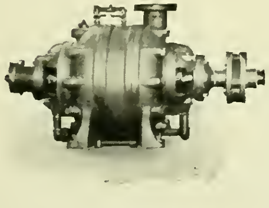
(National Standard Type A and B)

The water passages are made larger than in pumps hitherto built. In order that it may start instantly at all times, the working cams and water casings are made of *solid brass.* The shafts are very heavy, with liberal bearings. Fittings include a pressure valve, a relief gauge and cone, a priming pipe and valve, a starting valve, two to six hose valves, and a capacity plate.



POWER TRANSMISSION FRICTION GEARS

Our heavy duty Grooved Friction Gears are of substantial design throughout. The spring yoke press frames with angular split bearings ensure even distribution of pressure over the full face of both gears. Also angle of grooves gives high efficiency results. Approved and accepted by the Underwriters for fire pump service.



TURBINE AND CENTRIFUGAL PUMPS

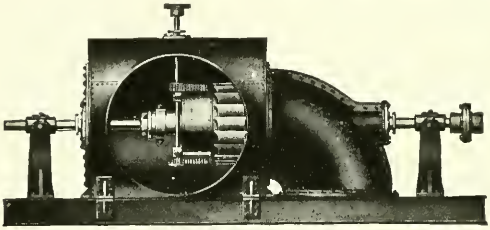
Our Turbine Pumps are especially adapted for operating by electric motors. Proper provision for every requirement for such pumps is made in our design. Absence from vibration readily tested by balancing a coin on casing when in operation.

Our Centrifugal Pumps are constructed with special reference to durability and long service, and are adapted for practically every requirement.

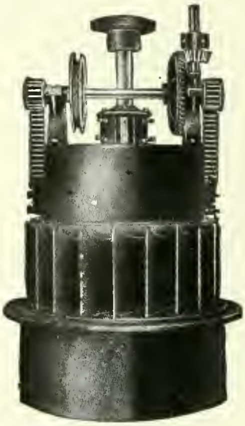
J. & W. JOLLY

HOLYOKE, MASS., U. S. A.

HORIZONTAL AND VERTICAL TURBINES



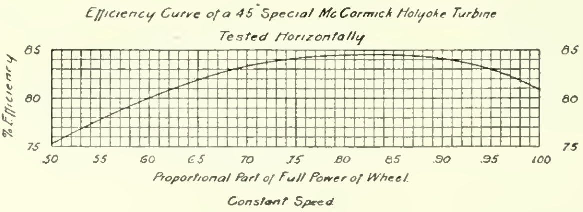
McCormick Cylinder Gate Turbines, Horizontal Type, in Steel Flume



McCormick Vertical Turbines

We build turbines in all sizes and styles, both cylinder and swing gates, to meet ordinary conditions, and design new wheels to meet special conditions of power and speed. We guarantee results in all cases. We would call particular attention to the efficiency curve shown below, which was plotted from results shown by the official test sheet. This wheel, when tested in the flume of the Holyoke Water Power Co., was mounted on a horizontal shaft, and discharged through a quarter-turn elbow and draft tube, thus giving results under actual working conditions.

Note that constant speed is maintained at all stages of power.



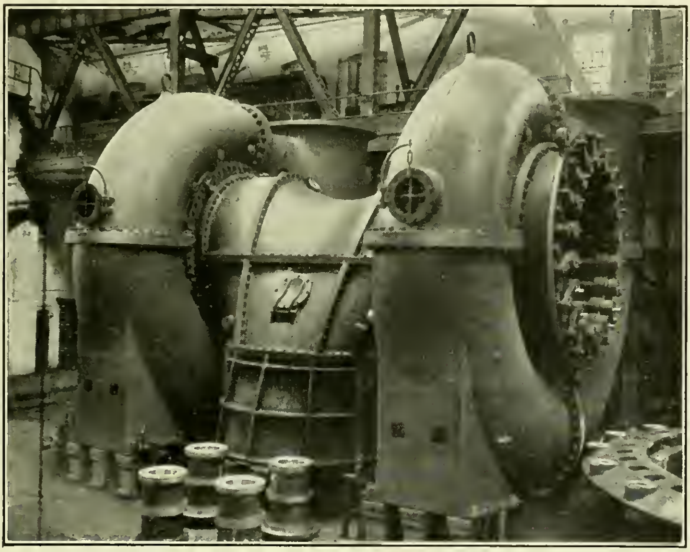
We also build Headgates complete with operating mechanisms of various designs for gates opening either vertically or horizontally.

I. P. MORRIS COMPANY

HYDRAULIC DEPARTMENT

PHILADELPHIA, PA.

SPECIALISTS IN THE DESIGN AND CONSTRUCTION OF HIGH CLASS, HIGH POWER AND HIGH EFFICIENCY HYDRAULIC TURBINES



22,500 H. P. TURBINE

Two units furnished. Head, 168 ft. Speed 200 R. P. M.
Most Powerful Water Turbines in existence

The I. P. Morris Company in 1851 built its first hydraulic turbines, which were used continuously by the Water Works Department of Philadelphia until superseded by the present filtration system, in 1908.

Since these first turbines were built, the art of turbine design has reached the perfection shown by the accompanying extract from the Report of the Holyoke Water Power Company, on an experimental runner tested in April, 1912. This runner holds the record for efficiency at the Holyoke testing flume.

Gate	Head	R.P.M.	Horse-power	Efficiency
4	17.52	232	102.53	79.41
3.5	17.58	229.75	101.54	84.43
3.0	17.40	232	94.1	88.01
2.65	17.44	233.75	90.57	90.62
2.5	17.48	225.25	84.51	89.61
2.00	17.75	233.33	61.27	84.98
1.25	18.32	227.75	34.47	73.05

Since the I. P. Morris Company began active construction of hydraulic turbines, it has designed and built turbines aggregating 1,148,725 horse-power, of which 273,725 horse-power has been contracted for within the past eleven months.

Inquiries for turbines requiring special design will be given every attention.

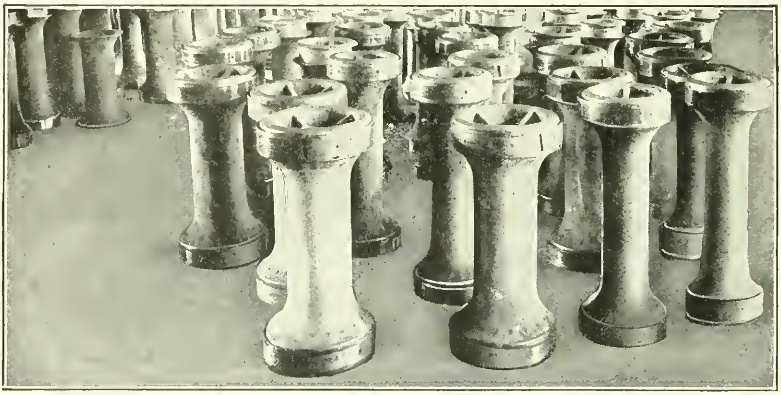
AMERICAN BALANCE VALVE CO.

JERSEY SHORE, PENNA.

BALANCED VALVE SPECIALISTS

Since 1890

DISTRIBUTION VALVES

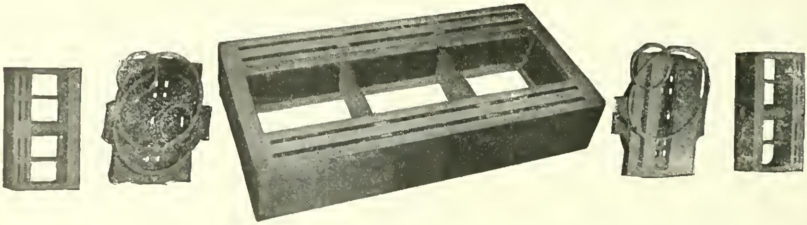


SEMI-PLUG Piston Valves for Superheated Steam or Saturated Steam of any Pressure up to 1000 Pounds.

These Valves are Frictionless, are Steam-tight and *REMAIN SO*. They are Maintained by DUPLICATE parts from *STOCK*. When under Pressure this Valve is a PLUG and when without Pressure it is a Snap *Ring* Valve.

Can be fitted to any Piston Valve Engine.

The Jack Wilson High Pressure Double-Ported Slide Valve for Pressures up to 240 lbs. and Superheat to 600° F.



BALANCED in all positions of travel. Double Admission, Double Exhaust and made for Internal or External Admission.

Can be fitted to any Slide Valve Engine.

When Designing or Repairing Engines, you should investigate these Modern Balanced Valves.

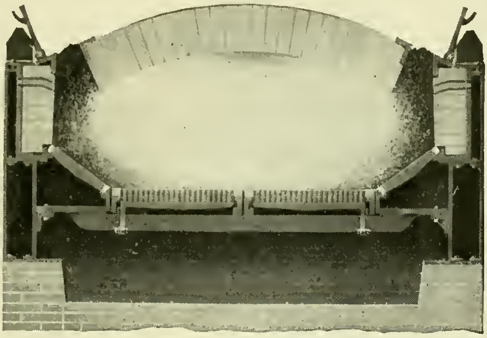
THE BURKE FURNACE COMPANY

CHICAGO, ILL.

APPARATUS FOR SMOKELESS COMBUSTION OF BITUMINOUS FUELS

Easy
To Operate

No
Complicated
Mechanism
to
Get Out
of
Order



Adapted
to
all kinds
of
Boilers
Both
Old and New

Cross Section Through Furnace

THE BURKE FURNACE

The Burke Furnace is of the gravity feed type and consists of a basket grate covered by a fire brick arch. Coal is fed to the feed pockets at the sides of the arch, cokes on the slanting side grates and when more fuel is needed moves toward the center grates, the fresh fuel taking its place on the coking grates.

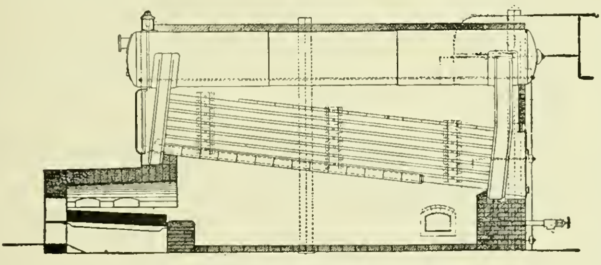
The coking process being continuous, the volatile gases (the smoke makers) are distilled gradually and therefore in such small quantity at a time that it is possible to supply the amount of air needed for combustion. The necessary high temperature is maintained by means of the fire brick arch. As the coked coal is moved toward the center of the furnace the green coal falls down below and behind it, so that the gases liberated from the fresh coal must practically pass through a bed of incandescent fuel, wherein they are heated, mixed with the air coming up through the grates, and burned without smoke and with very high efficiency.

Actual trials have shown a saving as high as 35% over ordinary settings.

The great losses due to inrush of cold air when doors are opened for firing are avoided in the Burke Furnace, as no fuel is fed through the front doors. These are for cleaning purposes only. As it is thus possible to maintain a practically constant temperature in the boiler setting, the enormous wear and tear on boiler and brickwork, due to expansion and contraction with changing temperatures, is successfully eliminated.

The cost of installation and operation is much lower than that of any mechanical stoker, yet the results obtained will stand comparison with the best on the market.

The Burke Furnace is no experiment; hundreds are in daily use throughout the United States and Canada. Send for catalogue.



Water Tube Boiler with Burke Furnace
A Typical Burke Furnace Installation

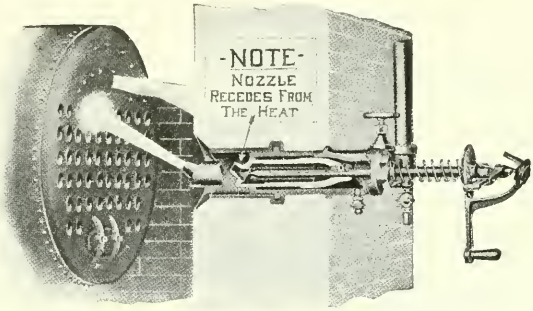
DIAMOND POWER SPECIALTY CO.

1534 Monroe Building
CHICAGO

Main Office and Factory
70 First St., DETROIT

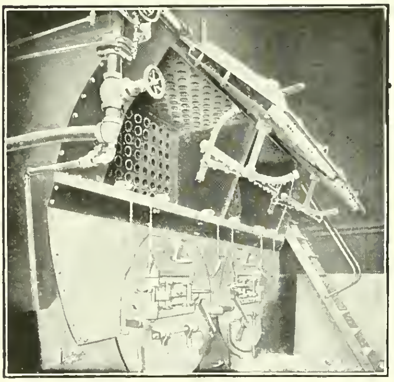
722 Singer Building
NEW YORK CITY

SOOT BLOWERS FOR ALL STANDARD BOILERS



1. Rear-End Design—For Return Tubular Boilers

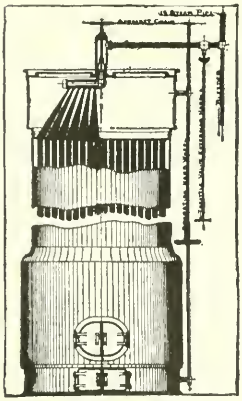
—have been developed through 15 years of experience in this field. These Diamond Soot Blowing Systems are permanently attached to the boiler—enabling the engineer to keep his heating surfaces free from soot at all times, by a simple mechanical steam-scouring action.



2. Front-End Design—For Fire Tube Boilers

OVERCOMING SOOT

The wasteful action of soot deposits—both in lost heat and in boiler depreciation — make necessary a positive method of overcoming this. In meeting this need these Soot Blowers produce an important economy in the plant.



3. For "Manning" Boilers

CLEANING FIRE TUBE BOILERS

Horizontal Return Tubular Boilers are cleaned from either rear or front end. The "Diamond" "Single Telescopic Jet" (see Fig. 1), "Swinging Arm," and "Revolving 5-Hole" Designs—are Rear End Designs. Fig. 2 shows the "Front End" Blower—open for inspection.

Vertical Fire Tube Boilers—"Manning" Type—are cleaned from top as shown in Fig. 3. Either "Swinging Arm" or "Telescopic Jet" designs are applied here.

DIAMOND POWER SPECIALTY CO.

Bourse Bldg.
PHILADELPHIA

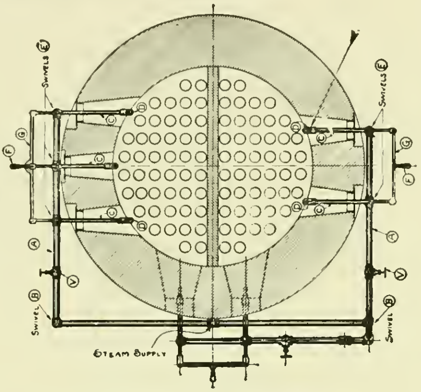
422 Fulton Bldg.
PITTSBURGH

Suite 4, 19 Pearl St.
BOSTON

SOOT BLOWERS FOR ALL STANDARD BOILERS

CLEANING WATER TUBE BOILERS

Soot is cleaned from the almost inaccessible points in many designs of water tube boilers—by special "Diamond" Blowers. These are now being used in a wide range of plants on such types of boilers as the following: "Hollow Stay Bolt," "Wickes" (see Fig. 4), "Stirling" (see Fig. 5), and Babcock and Wilcox.

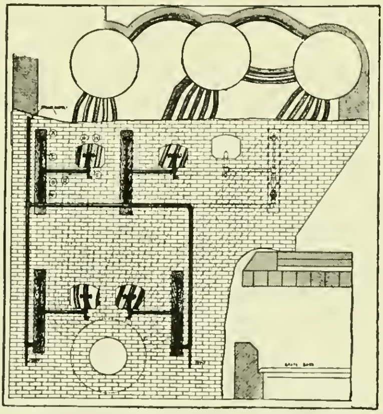


4. Cross-section showing application to "Wickes" Boilers

OUR ENGINEERING SERVICE

From complete files of plans and specifications covering hundreds of makes of boilers, our Engineering Department is able to co-operate with you promptly in considering the proper Soot-Blowing System for any particular installation. You should take advantage of this.

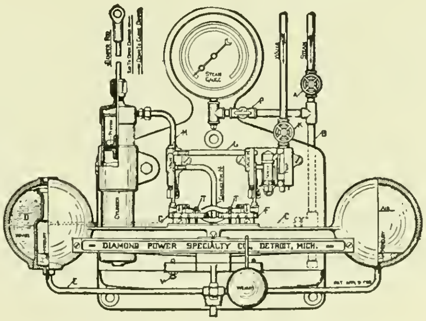
Send for detailed literature.



5. As applied to "Stirling" Boilers

DRAFT REGULATION

The regulation of the air supply in a furnace is automatically taken care of by the "Diamond" Damper Regulator which opens and closes the drafts as required. (See Fig. 6.)

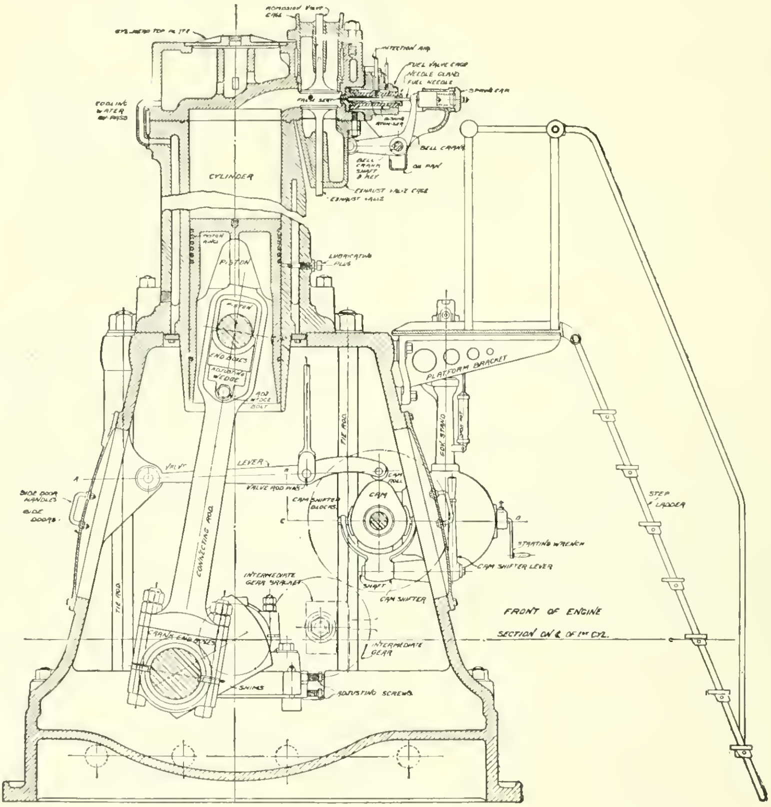


6. "Diamond" Damper Regulator

BUSCH-SULZER BROS.—DIESEL ENGINE COMPANY

ST. LOUIS, MO.

THE DIESEL ENGINE MANUFACTURERS



These engines are the most economical prime movers known.

Use crude or cheapest grade fuel oils or distillates.

Do not require heating up to start.

Have no electric or hot tube igniters.

Combustion gradual—no explosion; smooth running and evenness of speed thereby secured.

Fuel fed automatically from fuel tank into combustion chamber and amount injected governed by load on engine.

Start cold and pick up full load in a few minutes—stopped as readily—cut out stand-by losses.

None of the auxiliaries of steam or gas producer plants.

Operate in parallel alternating or direct current generators.

BUSCH-SULZER BROS.—DIESEL ENGINE COMPANY

ST. LOUIS, MO.

THE DIESEL ENGINE MANUFACTURERS

Our Diesel Engines produce power at lower cost than any known prime mover; for instance, will develop electric current at 1-2 cent or less per Kilowatt Hour; pump water at 2-10 of a cent per 1,000 Gallons at 50 pounds pressure and make ice at 18 cents a Ton.

Sold under the broadest kind of guarantee.

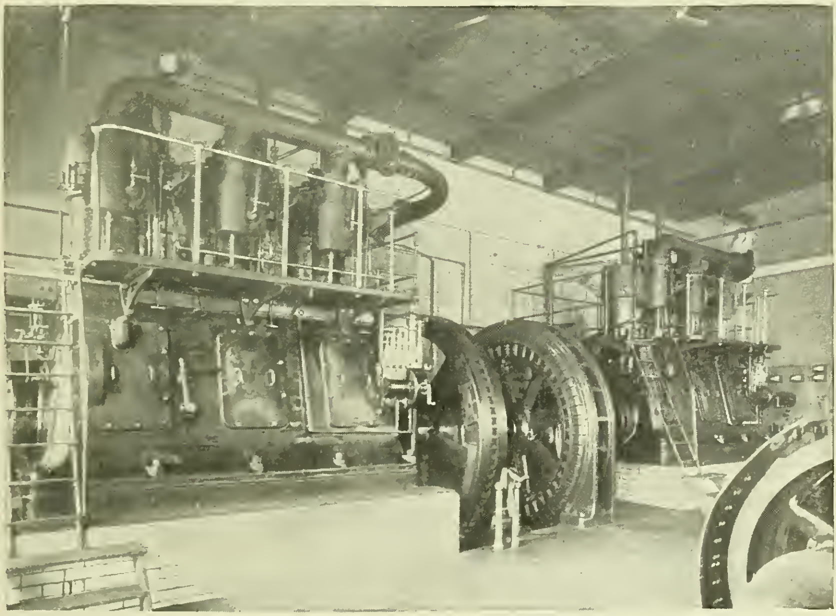
Economize space. No coal yards, bins, ash dumps or fuel handling space needed. Less building to keep up and pay taxes on or more space for factory, warehouse or shipping room.

No smokestacks—no smoke, soot or dirt—most clean and sanitary power plant built.

Over 100 satisfied users in the U. S. will tell you about the Busch-Sulzer Bros.—Diesel engine.

Write for details, cost of installation, operation and tests.

By agreement with Dr. Diesel this company has exclusive right to his inventions, improvements and advisory engineering services for the U. S. and Canada.

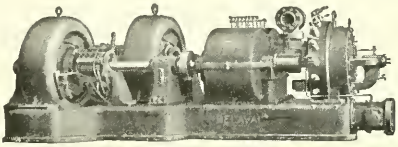


Diesel Engine Plant, Gorham Mfg. Co , Providence, R. I.

DE LAVAL STEAM TURBINE CO.

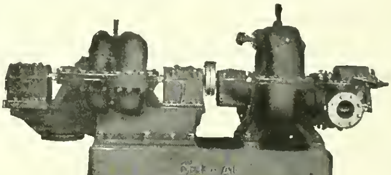
TRENTON, N. J.

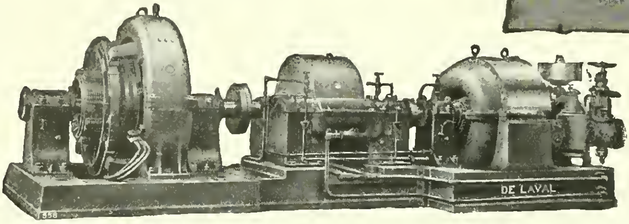
STEAM TURBINES, SINGLE AND MULTI-STAGE, FOR ALL SERVICES. CENTRIFUGAL PUMPS, SINGLE-STAGE AND MULTI-STAGE, FOR ALL CAPACITIES AND ALL HEADS. CENTRIFUGAL BLOWERS AND AIR COMPRESSORS, AND SPECIAL CENTRIFUGAL MACHINERY.



De Laval Steam Turbines are built for direct connection to alternators and other high speed machinery, and also in connection with the De Laval Speed Reducing Gear for driving standard-speed, direct-current generators, centrifugal pumps, centrifugal blowers, and for rope and belt drives; also for driving paper machines, rolling mills, hoists and other slow and moderate speed machinery.

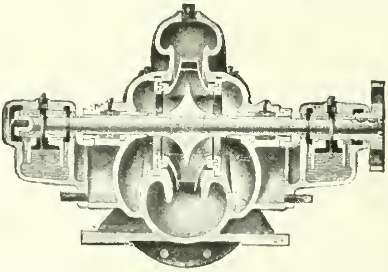
The De Laval Single-stage Steam Turbine is the most efficient of all turbines for the range of sizes from 5 H.P. to 600 H.P. One and the same turbine can be fitted with nozzles for any steam conditions. Send for Catalog A.



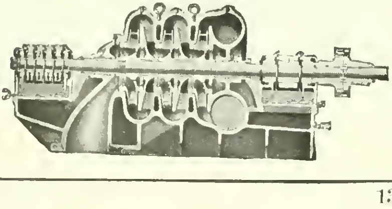


The De Laval Multi-velocity-staged Steam Turbine gives high efficiency in non-condensing, back pressure or low pressure service. By reason of the multiple stages, it may be applied without gears to the direct driving of small direct-current generators, centrifugal pumps and blowers, etc. All internal parts are accessible upon lifting the casing cover, and without disconnecting steam or exhaust piping. All wearing parts, such as blades, guide vanes and nozzles, are easily replaced at small expense. Send for Catalog F.

De Laval Multi-stage Turbines are of the "multi-cellular" type, consisting of a series of De Laval Wheels in separate chambers. Through using the proper turbine speeds, made possible by the gear, the number of stages, the length of shaft and complexity have been minimized. The wheels are enclosed in an impenetrable steel armor. These turbines are built in all sizes above 500 H.P. Write for Catalog D.



De Laval Centrifugal Pumps are built in both single-stage and multi-stage types and with horizontal and vertical shafts and in all capacities and for all heads. De Laval Centrifugal Pumps are distinguished by the fact that all wearing parts are removable and replaceable, including bearings, shaft sleeves, impellers, labyrinth packing rings, etc. All internal parts may be removed upon lifting the casing cover, and without disconnecting suction or discharge piping. Send for Catalog B.



THE VEEDER MANUFACTURING CO.

HARTFORD, CONN.

MAKERS OF CYCLOMETERS, ODOMETERS, TACHOMETERS, TACHODOMETERS, COUNTERS AND FINE DIE CASTINGS.

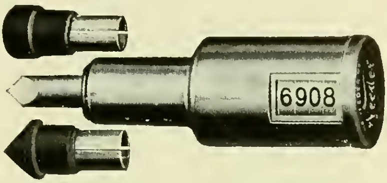
COUNTERS AND TACHOMETERS

Absolutely reliable instruments for the indication of speed or the recording of output from machines.

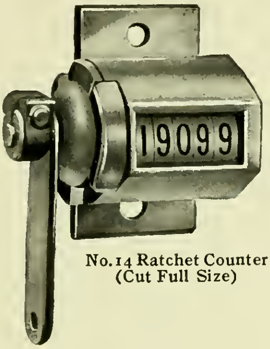
Essential in Scientific Management for the Piece Work Records.

ELEVATOR MILEAGE

Recorders which we furnish make it possible to keep satisfactory records of service and upkeep.



Veeder Clutch Speed Counter



No. 14 Ratchet Counter
(Cut Full Size)

Counters, both ratchet and revolution, for practically every purpose. Especially suitable for use on Punch Presses, Voting Machines, Cash Registers, Slot Machines, Railway Semaphores, Neo-styles, Telephone Pay Stations and Automatic Machinery. We illustrate only two of our many styles.

VEEDER FORM C TACHOMETERS

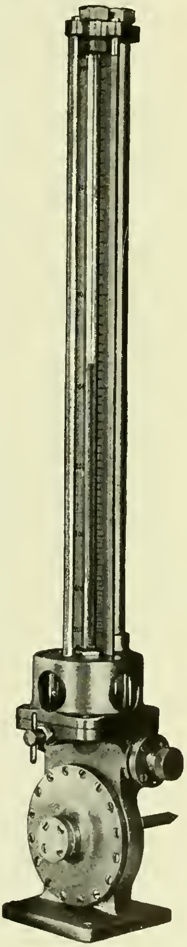
The Form C Tachometer is a portable instrument for indicating revolutions per minute of dynamos, motors, shafting, etc. The only moving part is the paddle wheel in the centrifugal pump. This has radial blades and hence indicates when run in either direction. When paddle is revolved, the liquid is forced out of pump and up the indicating tube by centrifugal force.

No. 4 Revolution Counter
(Cut Full Size)

FINE DIE CASTINGS

When accurately made parts are wanted in large quantities, they may be manufactured most economically by casting to their finished size and shape. Our alloy has a tensile strength of about 15,000 lbs. and compressive strength of 20,000 lbs. It is about equal in strength to aluminum or cast iron.

Complete Catalogue on request.

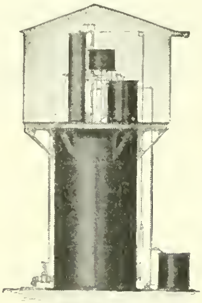


Form C Tachometer

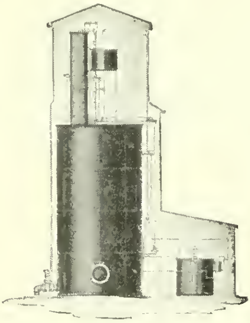
WM. GRAVER TANK WORKS

209 La Salle St., CHICAGO, ILL.

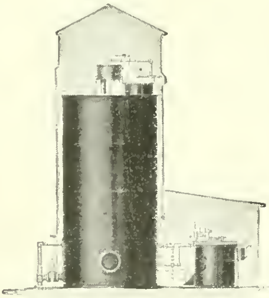
BARTLETT-GRAVER WATER SOFTENER AND PURIFIER FOR GENERAL BOILER, INDUSTRIAL AND DOMESTIC PURPOSES. STEEL TANKS AND PLATE WORK



Type AG.



Type BW.



Type B.

In softening and purifying water the BARTLETT-GRAVER WATER SOFTENER and PURIFIER uses the standard "Porter-Clark" process which consists of adding lime and soda ash to the hard water to precipitate the hardening ingredients. These precipitated solids are removed from the water by sedimentation and subsequent filtration.

The apparatus is automatic and continuous, the chemical being added to the water in proportion to the amount of water treated, the flow of the water being uninterrupted and the amount of water entering the system being the same as the amount of softened water withdrawn.

The chemicals are used in the form of suspension and are prepared once in twelve hours. They are accurately proportioned by means of the Bartlett-Graver Displacement System in which no mechanical device comes in contact with the chemicals. To this feature is due the accuracy of the apparatus.

The precipitated solids fall to the bottom of the settling tank from where they are periodically withdrawn through the sludge piping. A filter is located at the top of the settling tank to remove any precipitate which does not settle.

The softener is built with two types of filter depending upon the class of service demanded. Excelsior is used as a filtering medium in one type and crushed quartz in the other. The quartz filter is equipped with reverse air wash system which automatically removes the accumulated sludge.

The apparatus is built in four types, the difference between them being the position of the chemical tank and the source of power. In type "A" the power is derived from a water wheel operated by the incoming water, the chemical tank being located at the top of the apparatus. Type "AG" is the same with the exception of a second chemical tank being located upon the ground level. In type "BW" the chemical tank is located upon the ground level, all power for agitating and pumping the chemical being derived from an over-shot water wheel which is connected by vertical shaft to the lower operating parts. In type "B" the chemical tank is located upon the ground level, the power necessary to drive the apparatus being derived from either an electric motor or small engine.

A complete catalogue thoroughly describing this apparatus will be mailed upon request.

FETTA WATER SOFTENER CO.

RICHMOND, INDIANA

THE FETTA SYSTEM OF WATER SOFTENING

FOR POWER PLANTS, LAUNDRIES AND DOMESTIC USE

This system consists of our regular equipment of tank, filters, economizer, stick chemical feeder and feed water control. We also show in the illustration a feed water heater attached. Where no steam is available, this system also treats water cold. There are, however, many decided advantages in the hot process.

The distinctive and superior features are:

(a) *The Stick Chemical Distributor.*

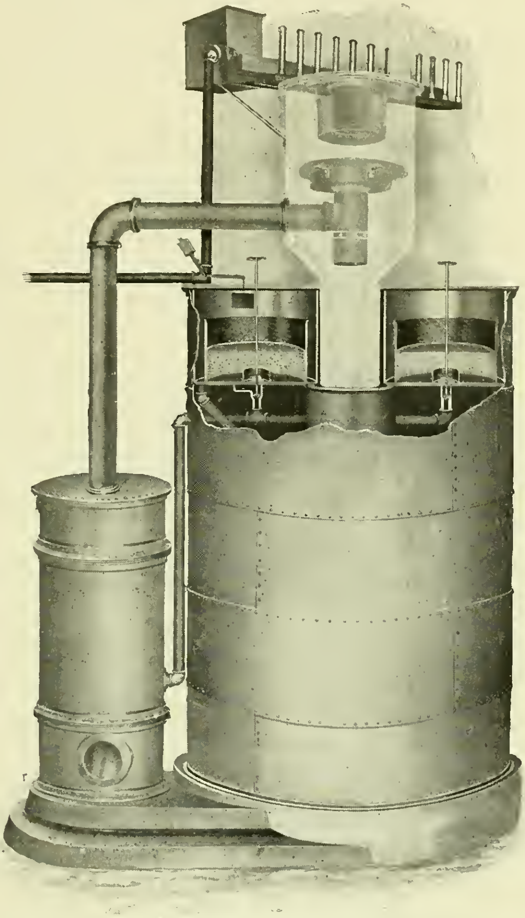
(b) *The Well Arranged Precipitating Tanks.*

(c) *The Permanent Quartz Sand Filters.*

(d) *The Fetta Feed Water Control.*

By this arrangement we produce soft water with the least possible amount of chemicals and deliver the water at temperatures as high as 180° F. This means *economy in size of plant, amount of chemical, and fuel consumed.*

The supply of cold water is intermittent, thus making possible a proper settling of precipitated impurities, but the delivery of soft water is continuous. This system therefore combines the advantages of the "Intermittent" and "Continuous" systems.



The Fetta Economizer System

It is an established fact that scale forming salts in hard water are more readily acted upon by chemicals, and thrown down, when the water is at a high temperature, than when treated cold. This means that the action is more rapid and complete, and that the treating can be made much smaller than would otherwise be possible, at the same time giving better results. All of these advantages are secured in the Fetta System.

Complete information regarding the Fetta System of Water Purification, and Bulletins describing special features mailed on request.

THE WM. B. PIERCE COMPANY

43 N. Division Street, BUFFALO, N. Y.

TUBE CLEANERS AND FLUE GAS ANALYZERS

THE DEAN BOILER TUBE CLEANER

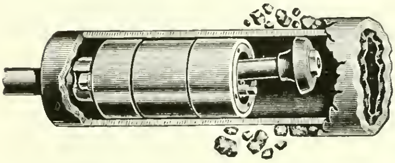


Fig. 1

For Removing Scale from Return Tubular Boilers

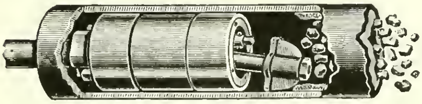


Fig. 2

For Removing Scale from Water Tube Boilers

The Dean is designed for removing scale from the tubes of water tube and return tubular boilers, locomotives, condensers and evaporators. For operating in the curved tubes of Stirling boilers and the Arch tubes of locomotives the cleaner is built shorter than the standard Dean and is barrel-shaped, thus permiting its passing around the bends.

The Dean is operated by air or steam and cleans from 10 to 30 tubes an hour. It removes scale by the well known principle of vibration. By a series of light, rapid taps, the force of which may be regulated regardless of the pressure employed, it loosens the scale from the tube. In return tubular boilers the scale falls off in showers. In water tube boilers the loosened scale is broken into chunks by the cutting edges of the vibrator (see Fig. 2) and driven out by the exhaust.

In cleaning the tubes of water tube boilers the Dean will prove far superior to any other means in thoroughness of work and durability. And besides removing the scale it will remove the hard baked soot, which cannot be removed by any other means.

The same cleaner by changing vibrators may be used in water tube and return tubular boilers. At a small additional cost extra attachments making the cleaner suitable for operating in two or more different sizes can be furnished.

The Dean is a tube inspector as well as a tube cleaner and discloses leaks, but *does not* produce them.

It also shows the fallacy of the use of the compounds.

For further information write for Booklet No. 130.

We also manufacture Gas Analyzers for determining CO2 O and CO.

Write for catalogue.

THE UNITED STATES GRAPHITE CO.

SAGINAW, MICH., U. S. A.

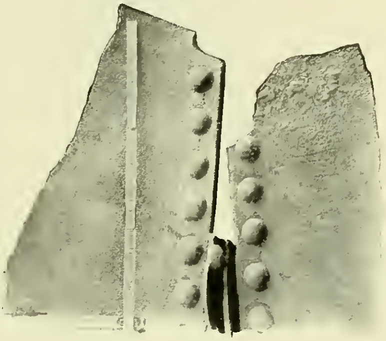
MINERS OF GRAPHITE AND MANUFACTURERS CF GRAPHITE PRODUCTS

U. S. G. CO.'S MEXICAN BOILER GRAPHITE

Loosens Old Scale by working through the minute cracks and checks (caused by the unequal expansion and contraction of metal and scale) and depositing itself on the inner surfaces of tubes and shell, with the result that the scale no longer adheres tenaciously and may be removed with comparative ease.

Prevents Formation of Hard Scale by its fine particles becoming distributed amongst the particles of scale. The graphite being a lubricant prevents the scale particles from adhering tightly to each other. The scale is thus kept in a loose condition and may be easily removed.

Has No Chemical or Electrical Effect because the action of graphite (which is carbon in its most inert form) is entirely MECHANICAL. This means that there will be no pitting of the boiler from chemical action or from electrolysis. Foaming cannot be caused by use of U S G CO'S MEXICAN BOILER GRAPHITE.



Sections of Old Scale Removed from a Feed Water Heater After Being Softened and Broken Down by U S G CO'S MEXICAN BOILER GRAPHITE.

Cannot Be Carried Over By the Steam.

Being insoluble U S G CO'S MEXICAN BOILER GRAPHITE cannot under normal conditions pass out of the boiler with the steam. For certain industrial processes, such as Brewing, Laundrys, Ice Plants, etc., this quality is very desirable and important.

Is Equally Suitable For All Feed Waters.

Since it acts MECHANICALLY instead of chemically, U S G CO'S MEXICAN BOILER GRAPHITE may be used successfully with any feed water. This does away with the expense of analyzing feed waters and keeping on hand a number of different compounds to suit different feed waters.

Only 14 lbs. of Soft Scale was found in a 300 h.p. Wickes Vertical Water Tube Boiler after Six Weeks' Operation without Cleaning. During this period U S G CO'S MEXICAN BOILER GRAPHITE was used regularly.

Not All Graphite Can Be Applied For This Purpose, because the graphite used must not only be a very pure form of carbon but also of a nature suitable for lubrication. All formations of graphite are not pure carbon, nor are they all good lubricants. Moreover, graphite to work successfully in a boiler should be ground to an unusual degree of fineness to insure its constant circulation through the boiler. Besides it must be very, very fine in order to work into and disintegrate old scale. U S G CO'S MEXICAN BOILER GRAPHITE is such a product. It has a record and a reputation.

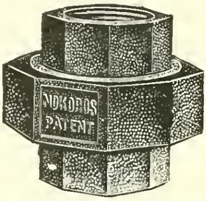
Put up in 100 lb. Kegs and Barrels (about 350 lbs.).

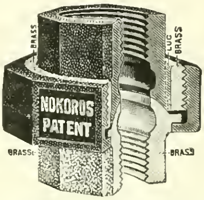
Send for Testimonials and Booklet "H."

ILLINOIS MALLEABLE IRON CO.

1801-25 DIVERSEY PARKWAY, CHICAGO, ILL.

THE NOKOROS PATENT UNION

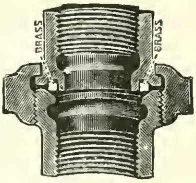




The only Union made absolutely NON-CORROSIVE at all contact points.
Non-corrosive Brass to Iron thread connection.
Non-corrosive Brass to Iron seat between ring and tail-piece.
Octagon shape, a monkey wrench will turn.

Size, inches..........	1/4	3/8	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3
Plain, each...........	.19	.22	.27	.40	.48	.66	.80	1.14	2.10	2.65
Galvanized, each.....	.23	.26	.34	.49	.60	.82	1.10	1.40	2.75	3.50

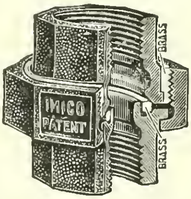
THE COMPRESSION DISC PATENT UNION

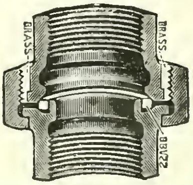


Heavy Pattern Air Furnace Malleable Iron with Brass Valve Seated Disc. The face of each threaded section is beveled to receive a brass disc, and the connecting up of the union COMPRESSES the brass DISC against the recesses, making a permanent steam metal seat.

Size, inches..........	1/4	3/8	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3
Plain, each..........	.30	.40	.50	.60	.80	1.20	1.60	2.00	3.20	4.80
Galvanized, each.....	.45	.60	.75	.90	1.20	1.80	2.40	3.00	4.80	6.20

THE NEW IMICO PATENT UNION





New Imico Unions are made of non-corrosive malleable iron, extra heavy, with bronze metal valve seated disc and non-corrosive ring connection.

Size, inches..........	1/4	3/8	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3
Plain, each...........	.30	.40	.50	.60	.80	1.20	1.60	2.00	3.20	4.80
Galvanized, each.....	.45	.60	.75	.90	1.20	1.80	2.40	3.00	4.80	7.40

Also Mfgr's of

MALLEABLE and CAST IRON FITTINGS

Write for Catalogue

THE KELLY & JONES CO.

GREENSBURG, PA.

Manufacturers of

BRASS AND IRON PIPE FITTINGS, BRASS AND IRON VALVES, COCKS, ETC.

FOR STEAM, GAS, WATER, AIR AND OIL

Cast Iron Fittings

CAST IRON FITTINGS

All styles, screwed or flanged for all pressures.

All K. & J. cast iron fittings are made of the best grade iron, threads cut true to standard gauge and each fitting recessed.

Malleable Fittings

"Excelsior" Valves
Brass—High Pressure

MALLEABLE FITTINGS

All styles—for all pressures. Plain, beaded or flat band.

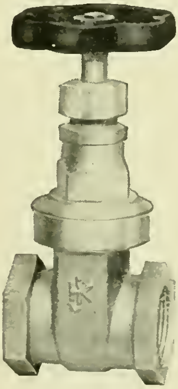
BRASS FITTINGS

Screwed or flanged, rough or finished — standard or extra heavy.

DRAINAGE FITTINGS

Our line of special recessed fittings for wrought iron drainage systems has been in satisfactory use for years.

Jenkins Disc Brass
Valves



Brass Wedge Gate
Valves

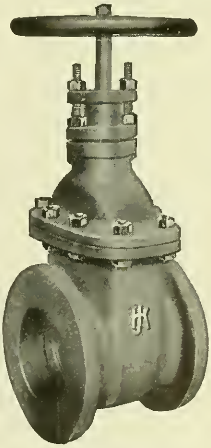
BRASS VALVES

We make a brass valve for every purpose and each valve a perfect product. Only the best grades of raw material used and each valve thoroughly tested to pressure recommended.

IRON BODY VALVES

Our line of iron body valves is most complete. We make every style and for all pressures.

SEND FOR COMPLETE CATALOG



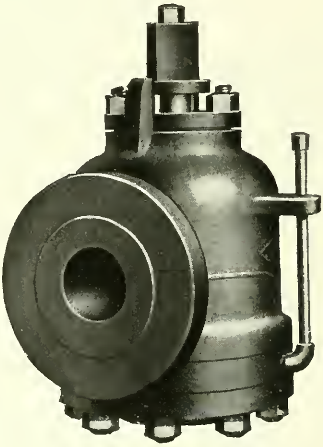
Iron Body Wedge Gate
Valves

JOHN SIMMONS COMPANY

104-110 CENTRE STREET, NEW YORK, N. Y.

IRON PIPE, FITTINGS AND VALVES

ROTHCHILD ROTARY GATE VALVES



External View

The severe conditions governing the successful operation of the high-pressure steam plant of today demand valves that will operate *quickly* and *positively* will not *leak* or *stick* and will stand up under the hardest strains.

That's where the Rothchild Rotary Gate Valve comes in!

It is a *Hard Service* Valve any way you take it, especially as a boiler Blow-off. It is so radically different from other valves that it is in a class all by itself. Take a good look at the accompanying cut and see for yourself.

First of all note the simplicity—The gates are entirely surrounded by uniform pressure and automatically take up their own wear by *pressure alone*—making a perfect seat that simply *cannot leak* or *stick*.

There is no undue strain on the seating parts and the gates are held in the best position when fully open or closed, by positive stops—*in view of the operator*.

There are *many* other good points about the Rothchild that we will not attempt to tell here. Let us send you our booklet describing it in detail. Equally good for Steam, Water, Ammonia, Gas, Air, Oil or other Fluids. Hot or Cold.

Send for our general catalogue, showing full line of pipe, fittings, tools, valves and specialties for the steam and plumbing supply.

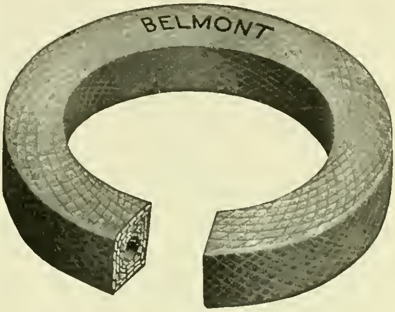
DIMENSIONS AND WEIGHTS OF ROTHCHILD ROTARY GATE VALVES

Size	Face of Inlet to Centre of Valve	Face of Outlet to Centre of Valve	Weight
1¼ inch	2⅛ inch	2⅛ inch	6¼ lbs.
1½ "	2⅝ "	2⅝ "	12¼ "
2 "	3⅜ "	3⅛ "	23 "
2½ "	5¾ "	4⅛ "	83¾ "
3 "	6¾ "	4⅞ "	123 "

CLEMENT RESTEIN COMPANY, INC.

PHILADELPHIA, U. S. A.

BELMONT HOLLOW CENTER PACKING



Belmont Hollow Center Packing, Style No. 19

This is a recently perfected steam and hydraulic packing, which is made with a "Hollow Center," for, while the hole weighs nothing and therefore costs nothing, this hole is really something of great importance.

Belmont Hollow Center Packing, Style No. 19, a ring of which is here shown, has several important advantages over all solid packings of either the square, round or wedge construction. In removing the old style "rubber core" and making this material with a hollow center the manufacturers have achieved remarkable results. First, having a hollow center, it weighs much less than the solid packings, and price per pound being the same, the cost for packing a given rod or plunger is much less. More important than this, however, is the fact that the hole provides a place for all excessive expansion or swelling, with the result that there positively can be no more friction on the rod or plunger than is necessary to withstand the initial pressure. Nor can it become tight enough on the rod or plunger at any time either from undue gland pressure or swelling to run hot and "burn."

The hollow center packing is in a way automatic in its action under the changing conditions and does not require constant and careful attention to make it last long and give satisfaction. The secret of its success lies in the hollow space which provides for all conditions of both expansion and contraction. For example, when a solid or wedge shape packing begins to swell and expand from absorption of steam or hot water, extreme temperature or for any other reason, it immediately becomes tight on the rod or plunger, causing excessive friction which quickly wears away the packing. At this point, if the expansion is not entirely out and the Engineer neglects to loosen the gland nuts, its continued swelling is quite apt to cause the packing to run hot and burn.

The action of the Hollow Center packing under these conditions is quite different. The hole provides a place for this excessive swelling or expansion and instead of becoming tight on the rod or plunger, the swelling goes to the hole or point of least resistance. Result—no excessive friction—packing does not wear away quickly—cannot become hot and burn.

It is recommended for steam, ammonia and oils, for elevator plungers, hot and cold water pumps and all conditions of hydraulic service where a stuffing-box packing is required. It is particularly efficient on outside packed boiler-feed pumps for hot water where much trouble is experienced with solid packings on account of excessive swellings. Try it. Your initial trial order will be filled with a positive guarantee to give you satisfaction or no pay. May we have a trial order under these conditions? Booklet illustrating and fully describing Belmont Hollow Center Packing for the asking.

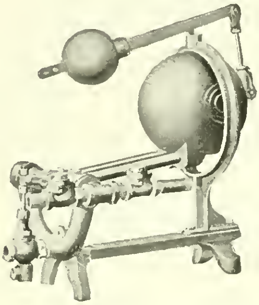
NASHUA MACHINE COMPANY

NASHUA, N. H. BOSTON, MASS.

NEW BUNDY STEAM TRAPS

Manufactured under Littlefield Patents

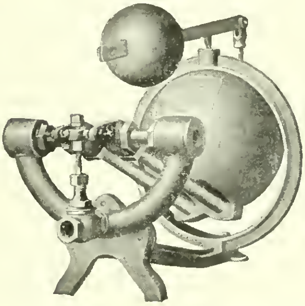
120 SERIES, NEW BUNDY RETURN TRAP



Used to automatically feed boilers, to return condensation to boilers, to relieve steam mains, heating coils, etc., to protect property, and to increase the efficiency of steam using apparatus.

Works up to 175 lbs. pressure.

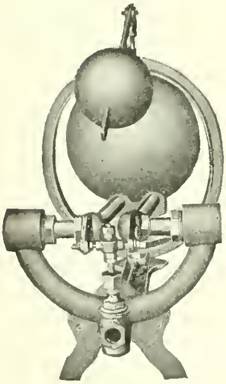
30 SERIES, NEW BUNDY SEPARATING OR TANK TRAP



New Bundy Separating or Tank Traps are designed to separate water from steam and are used for draining separators, headers, cylinder jackets, cooking apparatus, steam coils, etc. Discharging direct to the atmosphere or into hot well, heater, or tank.

Works up to 140 lbs. pressure.

90 SERIES, NEW BUNDY SEPARATING OR TANK TRAP



In this series we produce a machine of superlative quality. No amount of expense has been spared in our efforts to place this trap at the head of the class as a high pressure tank trap and one that can be offered to the purchaser under a positive guarantee covering both strength and efficiency and the performance of work under conditions of the hardest possible kind.

Works up to 250 lbs. pressure.

We also manufacture Vacuum Traps and Lifting Pump Traps

NASHUA MACHINE COMPANY

120 SERIES, NEW BUNDY RETURN STEAM TRAPS

No.	Pipe Connections		Lbs. of Water Delivered to Boiler per hr., based on 60 Operations, Normal Capacity	Drainage Capacity in Feet of 1 in. Pipe	Shipping Weight
	Water Inlet and Outlet, Inches	Steam Inlet, Inches			
120	¾	¾	800	4,500	160
121	1	1	1,380	6,000	200
122	1¼	1¼	2,100	10,000	290
123	1½	1½	2,520	20,000	353
124	2	1½	4,080	40,000	544
125	2½	1½	8,520	56,000	973
126	3	2	15,000	75,000	1,412
128	4	2	42,000	100,000	3,200

30 SERIES, NEW BUNDY SEPARATING OR TANK TRAP

No.	Pipe Connections		Capacity in Lbs. of Water Discharged per Hour at 50 Lbs. Pressure	1-inch Pipe Drained Lineal Feet	Shipping Weight
	Inlet, Inches	Outlet, Inches			
31	⅜	⅜	2,520	3,500	107
32	½	½	3,540	7,500	133
33	¾	¾	5,460	15,000	192
34	1	1	9,120	25,000	240
35	1¼	1¼	13,200	32,000	292
36	1½	1½	16,500	40,000	350
37	2	2	27,240	60,000	587
38	2½	2½	39,600	90,000	905
39	3	3	49,500	120,000	1,475

90 SERIES, NEW BUNDY SEPARATING OR TANK TRAP

No.	Pipe Connections		Capacity in Lbs. of Water Discharged per Hour at 150 Lbs. Pressure	Shipping Weight
	Inlet, Inches	Outlet, Inches		
91	⅜	⅜	3,720	128
92	½	½	8,100	151
93	¾	¾	14,100	235
94	1	1	24,600	269
95	1¼	1¼	36,800	362

The 30 series Separating or Tank Trap will work under any pressure up to 140 lbs., and the above capacities are based on what they will do under *50 lbs. pressure*, with a condensation ratio of 6 to 8 oz. per lineal foot per hour. The 90 series will work up to 250 lbs. and their capacities are calculated on *150 lbs.* The quantity of water that may be handled by these traps will vary directly with the given capacities, according as the pressure may increase or decrease.

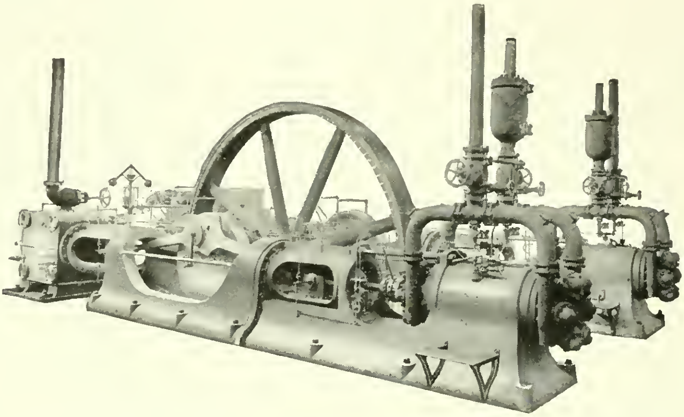
BUNDY TRAP RECEIVERS

No.	Length	Height	Diameter	No. of Inlets	Size of Inlets	Size of Outlets	Size of Blow Off
1	33 in.	10 in.	7½ in.	4	1 in.	1 in.	1 in.
2	33 "	11 "	8½ "	4	1¼ "	1¼ "	1 "
3	33 "	12 "	9½ "	4	1½ "	1½ "	1 "
4	33 "	13 "	10½ "	4	2 "	2 "	1 "
5	41 "	15 "	12½ "	4	2½ "	2½ "	1 "
6	42 "	18 "	15 "	4	3 "	3 "	1 "

THE VILTER MANUFACTURING CO.

MILWAUKEE, WIS.

CORLISS ENGINES, COMPLETE POWER PLANTS.
REFRIGERATING AND ICE MAKING MACHINERY.
BREWERS' MACHINERY, BOTTLERS' MACHINERY



Duplex Ammonia Compressor driven by Cross Compound Corliss Engine

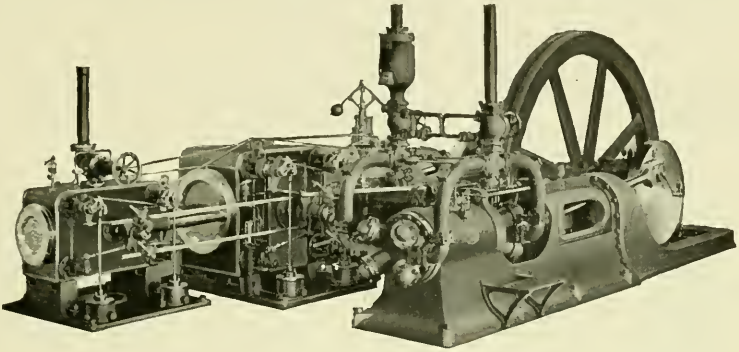
CORLISS ENGINES

Vilter Corliss Engines are equipped with an improved releasing gear giving high economy in steam consumption and reducing the wear on the reciprocating parts. They are of the horizontal type, built in all sizes from 25 H.P. up and furnished with either Girder, Heavy Duty or Rolling Mill Type Frames to suit individual load conditions and steam pressures. They are either Simple, Tandem or Cross Compound, condensing or non-condensing.

Excellent results are obtained where our Corliss Engines are used to drive our double-acting ammonia compressors. They are also admirably adapted for driving air compressors. For driving direct connected electric generators at higher rotative speeds the only change necessary is to equip them with our high speed, close-regulating governors.

All the refinements necessary to meet the exacting demands of modern steam engineering practice in a prime mover of this kind are incorporated in our designs. Steam and exhaust valves are double-parted and our cylinder castings are constructed with a single passage for the admission and exhaust at each valve. Cylinder clearance is reduced to a minimum. The steam and exhaust valve mechanisms are driven by either single or double eccentrics, the latter permitting a longer range of cut-off and greater overload capacity. Every part of our engine is designed with the idea of improving its mechanical efficiency and of giving the highest economy in every day operation.

THE VILTER MANUFACTURING CO.



Ammonia Compressor driven by Tandem Compound Corliss Engine

AMMONIA COMPRESSORS

Our ammonia compressors of the horizontal double-acting type are built in consecutive cylinder sizes ranging from 6″ x 12″ to 24″ x 48″. Sizes 13″ x 26″ and larger are also built in duplex units of from 125 tons to 750 tons refrigerating capacity. All sizes are steam driven by our direct connected Corliss Engines. They are also adapted for driving by belt or rope transmission.

The design of our compressors is remarkably free from complication and the construction is such that long life, satisfactory service and freedom from trouble is assured. Careful attention is given to exactness of workmanship during the building of our compressors, and no detail of construction, however slight, passes inspection unless the workmanship is up to standard.

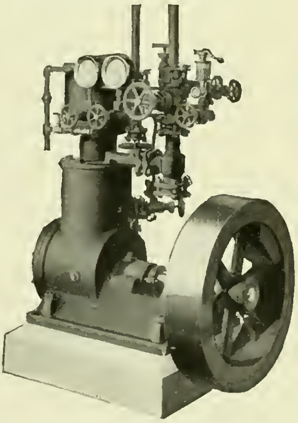
The cylinder and piston construction have special merit. The piston takes the form of an elongated sphere, presenting a large bearing surface to the cylinder walls. The cylinder heads are finished in conformity with this design and are turned to a true half sphere to conform to the shape of the piston, resulting in maximum strength and very small clearance spaces. This construction also gives the largest attainable valve area and permits the use of multiple suction and discharge valves in each cylinder head.

Every detail of our construction is fully described and illustrated in our bulletins, which we shall be glad to send to those interested.

SMALL BELT-DRIVEN SINGLE-ACTING COMPRESSORS

Our small vertical single-acting ammonia compressors are of the enclosed type, arranged for belt connection to an electric motor, gas or gasoline engine, line shaft, or small steam engine. They are designed with particular attention to the requirements of users of small ice-making and refrigerating plants. The fewest possible number of units and rigid construction insures perfect alignment and the least amount of wear on the moving parts. Particular attention has been given to the matter of automatic lubrication, accessibility for inspection and adjustment, and elimination of every troublesome element.

We are prepared to furnish this type of compressor in the following cylinder dimensions, viz: 4″ x 6″, 5″x 6″, 6″x 8″, 7″x 8″, 8″ x 9″ and 9″ x 9″ and to furnish full information relative to each size. These compressors are rated at from one to fifteen tons refrigerating capacity and are also furnished in duplex units with a single belt wheel.

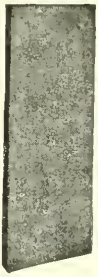


Vertical Enclosed Type Single-Acting Ammonia Compressor for Belt Drive

UNION FIBRE COMPANY

WINONA, MINNESOTA

WATER-PROOF LITH, UNION CORK BOARD, FOR INSULATION OF COLD STORAGE PLANTS, BREWERIES, ICE HOUSES, ETC.; LINOFELT FOR REFRIGERATOR CARS, ICE HOUSES, ETC.



WATER-PROOF LITH is an insulating board made of Patent Water-proof Rock Wool and Degummed Flax Fibres. It is furnished in the form of a board which can be handled and sawed like lumber. These boards are exactly 18″ x 48″ in area, and of any thickness from one-half inch to three inches.

It is claimed that, as compared with other insulating materials, Water-proof Lith offers important advantages as follows:

First —It has highest efficiency as a non-conductor.
Second —It is strong and easily handled.
Third —It will sustain considerable weight.
Fourth —It will not disintegrate
Fifth —It is uniform in composition.
Sixth —It will not absorb moisture and has no capillary attraction.

The usual method of applying Lith in brick or concrete buildings is to set up the boards either in one, or preferably two, layers, coating each board with a water-proof compound, and on the interior surface of the room covering the insulation with Portland Cement Plaster. Frequently each course is imbedded in Portland Cement Plaster against the wall in the same manner that tile are erected.

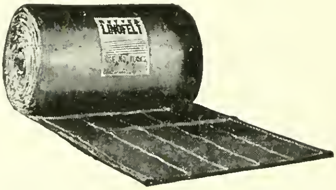
UNION CORK BOARD is composed of the best quality of pure granulated cork, each granule being treated at a high temperature and then instantly coated with a special water-proof mixture, then moulded into sheets and sawed to dimensions. Each granule of cork retains its full insulating value and each one is water-proof.

Union Cork Board is not burned in the process of manufacture, neither is there any compression applied to the boards beyond that which is necessary to make a close union of all particles.

Cork Boards are erected for the purposes of insulation just like Water-proof Lith Boards described above.

LINOFELT is a quilt insulation of degummed flax fibre manufactured by a special patent process from flax straw. The flax fibre is stitched between two thicknesses of 90-lb., 3-ply black waterproof paper to make the completed article, Linofelt.

FELTLINO is an insulating board made of selected flax fibre, strong, light of weight and high insulating value. The boards are 3 feet wide, 8 feet long and of any thickness from $\frac{1}{4}$ inch to $1\frac{1}{2}$ inches.



Linofelt and Feltlino are both widely used for insulating refrigerators. These two forms of insulation are adapted to ice houses and other wooden structures where low cost is essential.

The Union Fibre Company maintains a Construction Department and contracts to erect its insulation in any place. This Department has for the benefit of engineers data with regard to insulation and sketches of the different methods of application to overcome particular problems. Any of these will be sent to engineers free upon request. The company also issues booklets descriptive of its insulation, and furnishes samples upon request.

CHAMBERS BROTHERS COMPANY

Main Office and Works

FIFTY-SECOND AND MEDIA STS., PHILADELPHIA, PA.

BRICK MAKING MACHINERY

With capacities of from 10,000 to 100,000 daily

Briquetting Machines

Single and Double Shaft Mixers

The C. A. Wentworth Washing Machine

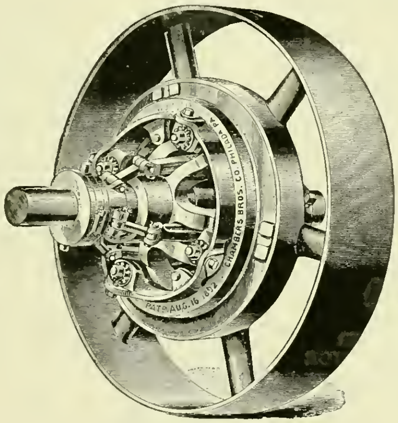
Roll Crushers

Disintegrators—Six Sizes

Pan Grinders for both Dry and Wet Grinding

7½	feet	diameter	Dry	Pan,	weight	25,000 lbs.
9	"	"	"	"	"	40,000 "
8½	"	"	Wet Pan,		"	46,000 "

Measuring and Feeding Machines, for feeding dry materials in fixed proportions.


CHAMBERS BROS. CO. PHILADA PA.
PAT'D AUG. 16 1892

The Flexible Friction Clutch Pulley

The Flexible Friction Clutch Pulley, a thoroughly good reliable disc clutch, unaffected by high speeds.

No. 4	Size	15	H.P.	per	100	Revs.	per M.
No. 5	"	22	"	"	"	"	" "
No. 6	"	30	"	"	"	"	" "
No. 7	"	40	"	"	"	"	" "
No. 10	"	95	"	"	"	"	" "

THE JEFFREY MANUFACTURING CO.

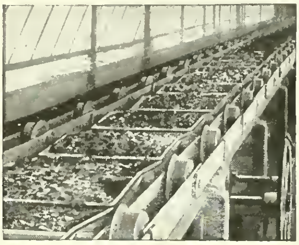
COLUMBUS, O.

New York
Boston
Montreal
Pittsburg

Atlanta, Ga.
Charleston, W. Va.
Birmingham
Chicago

St. Louis
Denver
Seattle

ELEVATING, CONVEYING AND POWER TRANSMISSION MACHINERY



JEFFREY OVERLAPPING-LIP PIVOTED BUCKET ELEVATOR-CONVEYER

for handling Coal, Ashes, Cement, Sand, Stone, Ore and other materials.

Designed and built to meet conditions.

Write for Catalog 32-A.

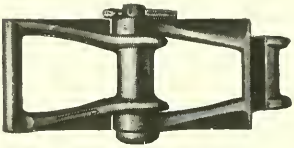
JEFFREY STORAGE BATTERY TRUCKS

save time, labor and expense in handling material in Industrial Plants.

No skilled labor required to care for battery or truck.

Battery will operate from one to two days on a single charge.

Bulletin No. 13-A and prices furnished on request.



JEFFREY CHAINS

are interchangeable with all other makes of Standard Link Chains of corresponding styles and numbers. Made of highest grade material, insuring uniform strength, pitch and surface finish.

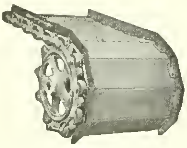
Special Chains made to order.

RUBBER BELT CONVEYOR

for handling all kinds of material.

Superior in design and construction.

Cost of installation, operation and upkeep is low.

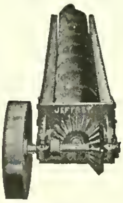


APRON CONVEYERS

made in sizes and weights to meet all requirements. Built with wood or steel slats.

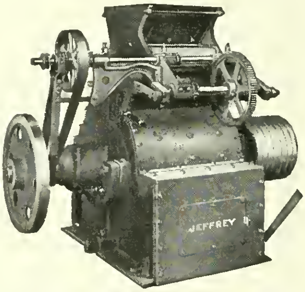
BUCKET ELEVATORS

for handling Coal, Stone, Ore, Sand, Gravel, etc. Made in styles and sizes to meet conditions.



SPIRAL CONVEYERS

for handling Sand, Cement, Coal, Stone, Ores, Grain, Seed, Fertilizer Materials, etc., are strong and reliable.

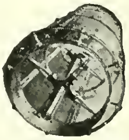


JEFFREY SWING HAMMER PULVERIZER

for reducing Limestone, Shale, Ores, Fireclay, Phosphate Rock, etc.

Has larger capacity, yields a more uniformly fine product, consumes less power and costs less for upkeep than any machine of its kind.

Request Catalog 31-C.



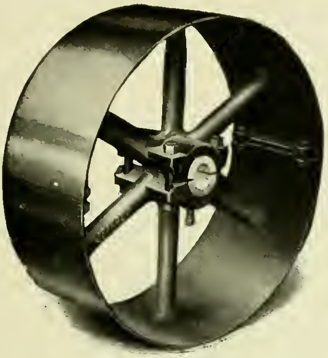
JEFFREY SCREENS

for treating Coal, Stone, Ore, Sand, Gravel, etc. Made in all sizes, Round or Hexagonal, of wire cloth or Perforated Plate.

Screen Catalog 69-B mailed on request.

THE JEFFREY MANUFACTURING CO.

Complete Line of Jeffrey Elevating, Conveying and Power Transmitting Appliances, as summarized below, are shown in the following Catalogs and Bulletins, which may be had upon request:



JEFFREY NEW IMPROVED SPLIT IRON PULLEY

with Interchangeable Bushings. Saves Time, Labor and Expense. Minimizes Power Loss.

JEFFREY TRAY ELEVATORS

for handling barrels, bags, boxes and miscellaneous packages.

Automatically load or discharge material on any floor of building.

JEFFREY ROPE DRIVE SYSTEMS

are correctly designed by Expert Engineers.

We build them in both American and English Systems.

JEFFREY PRODUCTS	Catalog Nos.
Bearings	50
Belt Conveyers	67-D
Belting—for conveyer duty	67-D
Belt Tighteners	50
Buckets, Elevator	43-A
Cable, Chain and Wire	33, 21
Cages, Mine	24
Car Hauls	33, 21
Cars	13
Chains—In Cast Steel, Forged Steel, Iron, Malleable Iron, Manganese Steel, for all elevating and conveying purposes	63, 64
Chutes	69-B
Clamps	
Clutches	50
Coal and Coke Handling Machinery	22, 42
Coal Crushers	31-C, 41
Coal Cutting Machines	18
Coal Drills, Electric, Rotary and Hand Power	16-A
Coal Pocket Equipments	25-A
Conveyers—for handling logs, mill refuse, lumber, coal, ashes, ore, sand, cement, gravel, phosphates, garbage, ice, miscellaneous freight and packages, bagasse, etc.	
Couplings, Clutch Couplings	50
Collars, Friction Clutches	50
Elevators—for all purposes	
Fans, Mine	26-A, 26-B
Floor Stands	50
Gears	50
Generators	17
Hangers	50, 51
Hoists, Crab, Electric Drum Hoists, Gantry Cranes	
Larries, Coke	13
Locomotives, Electric, Mine Haulage	15, 17
Locomotives, Storage Battery	13, 13-B
Picking Tables	42, 22
Pillow Blocks	50, 51
Power Houses	32, 32-A
Pulleys, moulded in metal or wood with single and double arms	50, 53
Rope-Transmission, Hoisting, Haulage	50
Screens, for any material, Rotary, Shaking or Vibrating	69-B, 69-D
Screw Conveyers and cast-iron end journal boxes for same	36
Shafting	50, 51
Sheaves	50
Spiral Conveyers	36
Sprocket Wheels	63, 64
Storage and Reclaiming Machinery	25-A
Tension Carriages	50
Trippers, Belt	35, 67-D
Wagon Loaders	25-A
Washeries, Coal	27
Weigh Hoppers	32, 32-A
Wire Cloth	69-B

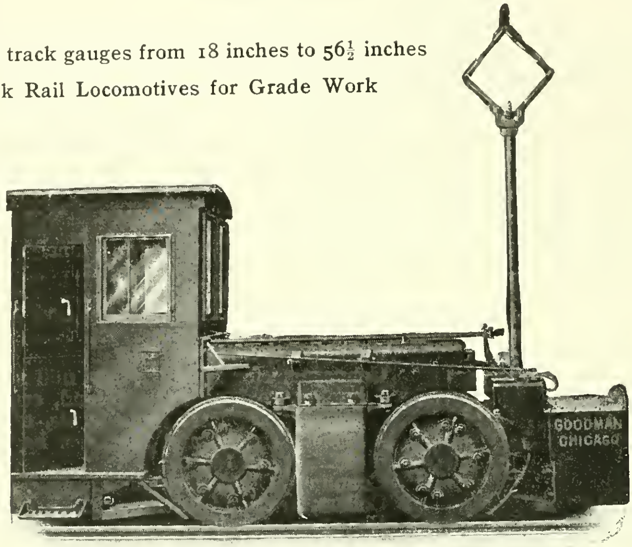
GOODMAN MANUFACTURING CO.

CHICAGO, ILL.

ELECTRIC AND GAS LOCOMOTIVES FOR MINE AND INDUSTRIAL HAULAGE

Built for track gauges from 18 inches to 56½ inches

Rack Rail Locomotives for Grade Work



Electric Locomotives—Single-Motor and Two-Motor in a great variety of types.

Gas Driven Locomotives—Using gasoline, alcohol or distillate.

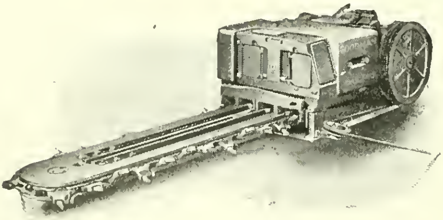
For detailed information, secure our bulletins:

No. 302—Single-Motor Locomotives	No. 351—Two-Motor Locomotives
No. 303—Industrial Locomotives	No. 502—Small types for switching, etc.

The **Goodman Handbook** also is of value as a compendium of information regarding locomotive practice and the Goodman Line. It will be sent free to persons interested in locomotives or coal cutting machinery.

We make a full line of

Coal and Clay Cutting Machinery



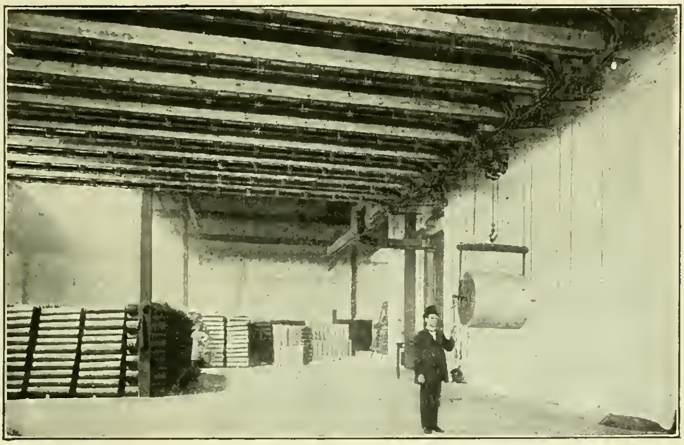
Breast Machines
Shortwall Machines
Longwall Machines

Ask for Bulletins
Nos. 101 and 103

RICHARDS-WILCOX MANUFACTURING CO.

AURORA, ILL.

OVERHEAD TROLLEY CARRYING SYSTEMS, AUTOMATIC FIRE DOOR FIXTURES; SLIDING DOOR HANGERS FOR WAREHOUSES, FACTORIES, ETC.

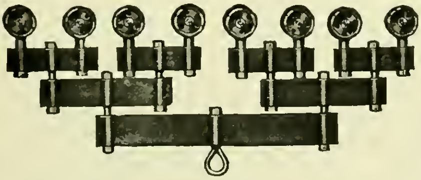


Typical Installation of Richards-Wilcox Overhead Trolley Track

The illustration above shows a typical arrangement of Richards-Wilcox Overhead Trolley Tracks in a large factory. The construction of this track is such that it is impossible for a carrier to drop out, and an ample factor of safety is provided against breakdowns.

Perfect flexibility is assured by a system of switches and turntables which provide for any desired arrangement of tracks.

These switches and turntables are absolutely "Fool-Proof" in that their construction leaves no possibility for carriers to drop no matter how turned or in what position.



Sixteen-wheel carrier, capacity 4,000 pounds, runs in No. 33 track. Eight-wheel carrier, capacity 2,000 pounds, runs in No. 33 track

CARRIERS

Carriers are built with hardened roller bearings, also with hardened ball bearings in five regular sizes having capacities up to 4000 lbs. as indicated by the table herewith, and can be run around curves of 24″ radius. A larger radius is advised, however, when it can be used.

Illustrated catalogs and full information concerning standard or special layouts and designs furnished on request.

PENNSYLVANIA CRUSHER CO.

STEPHEN GIRARD BLD'G.

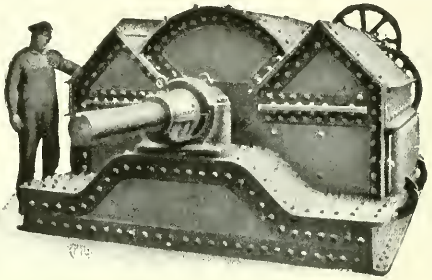
New York, 50 Church St. PHILADELPHIA, PA. Pittsburgh, Machesney Bld'g.

COAL CRUSHING & CLEANING MACHINERY FOR BY-PRODUCT COKE PLANTS, SWING-HAMMER CRUSHERS, BRADFORD COAL CLEANERS, ROTARY & JAW CRUSHERS, SINGLE ROLL CRUSHERS, GRINDING PANS, DELAMATER "SINK & FLOAT" TESTERS.

"PENNSYLVANIA" SWING-HAMMER CRUSHERS

Extensively used for pulverizing Bituminous Coals in By-Product and Bee-Hive Coking Plants; also for crushing Cement Rocks and Limestones in Cement Mills; also Lime, Shales, Bone and a multitude of other materials.

Main frame of fabricated Steel practically immune from breakage. Removable Steel Wear Liners, Ball and Socket Bearings, 6 and 8 rows of Hammers, large diameter Steel Discs, quick adjustable Grinding Cage. By weight the "Pennsylvania" is more than 90% Steel.



Hammer Crushers

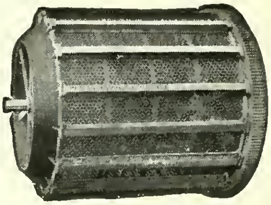
"PENNSYLVANIA" BRADFORD COAL CLEANERS

For Power Houses and By-Product Coke Plants

One of the important uses of this machine is to *automatically* remove impurities such as slate, bone, sulphur balls or binder from bituminous steam or coking coals, thereby reducing the ash and sulphur contents.

It is used extensively in preparing R.O.M. coals in By-Product Coking Plants and Bee-Hive Ovens.

In connection with its crushing and cleaning functions for R.O.M. coal for *large Power Houses*, the "Pennsylvania" Bradford is most efficient in removing stray iron, coupling pins, mine props and all sorts of impedimenta that damage Conveyors, Stokers and other Power House machinery.



Bradford Coal Cleaner

For Stoker feed it Crushes R.O.M. with less fines than Rolls, or any other type.

Absolutely automatic in operation, easy on horse power, runs 12 to 15 R.P.M., requires no labor to operate, other than occasional oiling. Is practically "fool-proof."

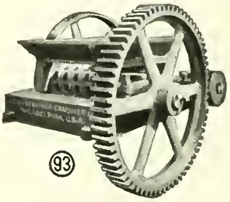
Several of these "Pennsylvania" Bradfords are successfully operating in connection with Coal Washers.

"PENNSYLVANIA" SINGLE ROLL CRUSHERS

Will take large pieces of Coal, Coke, Iron Ore, Limestone, Phosphate, Gypsum, Flint Clay and reduce to one inch and finer in one operation. A good machine for crushing coal for Stokers.

A single roll revolving close to a quickly adjustable breaker plate, in place of two rolls commonly used.

Roll is heavily back geared and rigidly held in its bearings to prevent spreading when feeding large lumps. This design is smoother running than the double Roll.



Roll Crusher

Bed frame in one piece and very rigid. Cut shows rear panel removed, exposing roll. Different designs of roll teeth are made. Moderate H. P., slow speed.

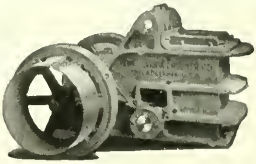
"PENNSYLVANIA" ROCK CRUSHERS

Powerfully built for *brutal* service, free from hot boxes, broken shafts and other common troubles.

Simple in design, of few parts, and require no expert for repairs. Thoroughly reliable under trying conditions.

Main frame is Open-Hearth Steel, shafts are high grade Steel forgings.

Bearing made to template, are kept in stock by operator and easily replaceable.



Rock Crusher

STURTEVANT MILL CO.

BOSTON, MASS.

NEW YORK PITTSBURGH CLEVELAND CHICAGO ATLANTA LONDON

CRUSHERS, CRUSHING ROLLS, GRINDING MILLS, LABORATORY MACHINERY, SCREENS, MIXERS, SACKING AND WEIGHING MACHINES

CRUSHERS—Coarse Crushers: for Breaking large Rocks and Ores to from 4 in. to 2 in. in size and finer.
Intermediate Crushers: for reducing to from 1 in. to 2 in. and finer.
Fine Crushers: for 1 in. to ½ in. crushing.
Rotar Fine Crushers: "Open Door" type for crushing soft and moderately hard rock.
Laboratory Crushers: for Sampling.

CRUSHING ROLLS—Balanced Rolls: Highest Grade, for Roughing and Finishing. Special Car-Box Bearings. Shocks only one quarter that of other Rolls. Springs back of each bearing.
Plain Balanced Rolls: One half the weight and approximately one half the price of Balanced Rolls.
Laboratory Rolls: for Sampling.

GRINDING MILLS—Ring-Roll Mills: for pulverizing dry materials such as Quartz, Cement Clinker, Coal, Limestone, Granite, Trap, Phosphate Rock, etc. Large outputs, small power and upkeep. Built in large and small sizes.
Emery Mills: for pulverizing soft and moderately hard materials, such as Lime, Gypsum, Talc, Clay, etc., etc. Horizontal and Vertical types—many sizes.
Sample Grinders: for Laboratory use.
Various other grinding devices for special purposes.

LABORATORY MACHINERY—Crushers: "Open Door" accessible type. Hand-Wheel adjustment—easily cleaned.
Rolls: for granulation and pulverization. "Open Door" type; automatic adjustments.
Grinders: for fine grinding—"Open Door," Hand-Wheel adjustment.
Screens: for accuracy, "Open Door."

SCREENS—Newaygo Separators; for screening everything screenable from 4 to 180 mesh. "Open Door" Inclined Vibrating type. Screen cloth automatically stretched and kept taut and vibrated by hundreds of little hammer blows upon its reenforced surface. Coarse Wire used for fine and accurate output. Large capacity.

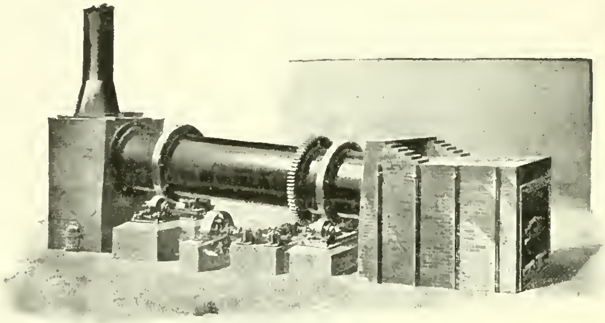
MIXERS—Hunso Automatic Mixers for Fertilizer.

SACKING AND WEIGHING MACHINES—Sawyer-Fulford Revolving Sacking and Weighing Machines. Accurate to an ounce. Sacks and weighs most anything that will go in a bag.

AMERICAN CLAY MACHINERY CO.

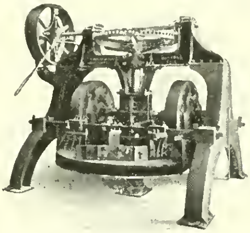
BUCYRUS, OHIO

ROTARY DRYERS, WET AND DRY GRINDING MILLS, ROLLER CRUSHERS, EXHAUST FANS AND DISC FANS

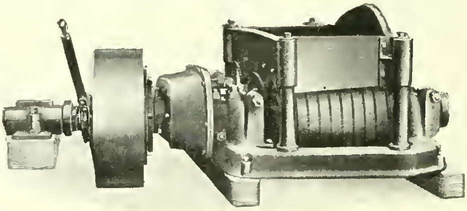


Rotary Dryer

ROTARY DRYERS. We manufacture a line of Rotary Dryers covering a large range of sizes and capacities and adapted to drying clays, shales, sands, marls, slurries, fertilizers, crushed limestone, and for calcining fire clays, roasting ores, and handling a great variety of other materials from which moisture must be removed at low cost, or in which special heat treatment is required.



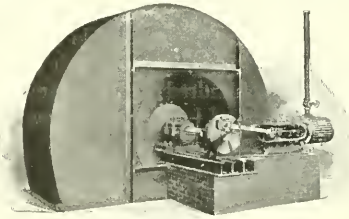
Grinding Mill



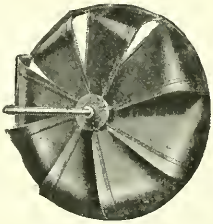
Roller Crusher

WET AND DRY GRINDING MILLS. We manufacture a line of wet and dry grinding machines, as shown in illustration. They are largely used by steel foundries in mixing and grinding silica sand and similar materials. They are made in a full range of sizes from 5 feet to 9 feet and for either motor or belt drive.

ROLLER CRUSHERS. We build a complete line of Roller Crushers for crushing and grinding clay, gravel, etc. They are extensively used for stone extracting where the material to be crushed contains stones.



Exhaust Fan



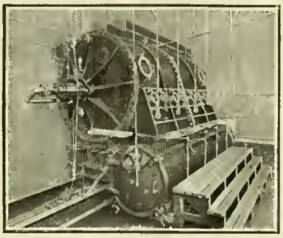
Disc Fan

EXHAUST FANS AND DISC FANS. We manufacture a complete line of Exhaust Fans including all sizes up to No. 360. These fans are made either three-quarter-housed or full-housed, and are extensively used for ventilation and dryer work. For full details see our No. 90 catalog.

J. P. DEVINE COMPANY

BUFFALO, N. Y.

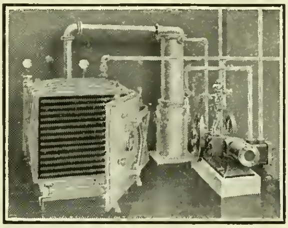
MANUFACTURERS OF VACUUM DRYING AND EVAPORATING APPARATUS



VACUUM DRUM DRYERS for Dyewood and Tanning Extracts, Milk and Food Products, Pastes, etc.

A rapid and uniform drying is effected because the drum dips into the solution and takes up a thin film of the wet material of 1–125 of an inch and less. The water is evaporated from the material at a temperature of from 117° F. to 96° F. according to the Vacuum in the apparatus of $26\frac{3}{4}''$ to $28\frac{1}{4}''$.

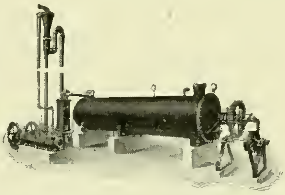
The drying process is continuous and independent of climatic conditions; free from dust and consequent elimination of danger to health of employees and destruction of property ; and at a minimum cost of operation, including labor.



VACUUM CHAMBER DRYERS for Colors, Dyes, Extracts, Salts, Rubber, Smokeless Powder and High Explosives, and other Chemical and Food Products.

The Vacuum Drying Chamber is designed to remove the water rapidly and at a low temperature from materials which cannot be dried by methods used heretofore without altering their chemical composition on account of their sensitiveness to heat. It may also be used with great saving in time, fuel, cost of plant, and working expenses for other substances where a low temperature is not an absolute necessity.

Materials which are difficult to dry in the atmosphere or which cannot be dried at all in the atmosphere without decomposition have all moisture removed from them in a very short time in the vacuum chamber without danger of impairing their qualities by overheating.



VACUUM ROTARY DRYERS, for Starch, Granular Substances, and Chemical By-Products.

The moist material is conveyed by an elevator into a hopper high above the manhole to facilitate the charging of the apparatus. After charging, the manhole is closed and a high vacuum produced by means of an air pump, the vapors passing into the condenser.

Concentric with the steam jacketed outside cylinder is a revolving inside drum, heated by live or exhaust steam, to which stirring blades are attached. The material to be dried is between the inside drum and the outside cylinder and is kept in constant motion by the stirring blades. Thus every particle comes into close contact periodically with the heating surfaces, and a very thorough and even drying process results.

VACUUM PUMPS of highest efficiency and of non-corrosive metals.

VACUUM PANS for any requirement and capacity in single or multiple effect.

CONDENSERS, ETC.

OVER 3,000 INSTALLATIONS IN DAILY OPERATION.

RUGGLES-COLES ENGINEERING CO.

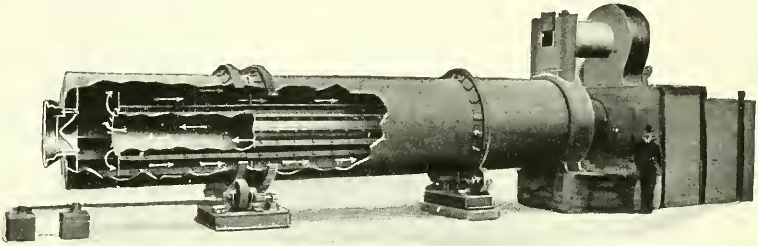
McCORMICK BLDG.
CHICAGO

50 CHURCH STREET
NEW YORK CITY

RUGGLES-COLES "Double Shell" DRYERS

Consulting and Contracting Engineers

Designers and Builders of Special Machinery



Section of Ruggles-Coles Dryer Showing Direction of Cases

RUGGLES-COLES "Double Shell" DRYERS

We build Ruggles-Coles "Double-Shell" Dryers for drying a large variety of inorganic and organic materials. For this work we have perfected six regular types of dryers, but for certain substances we build special dryers to order. After fourteen years' successful experience we know that the Ruggles-Coles Dryer is designed on the correct principle and point to more than 300 satisfactory installations in all parts of the world as evidence of its superiority.

Class "A" Dryer.—The principle of the Ruggles-Coles "Double-Shell" Dryer is that the material being dried passes the hot gases in the opposite direction to their travel. The Class "A" dryer consists of two concentric shells rigidly connected at the center. Between this point and each end are two sets of swinging arms allowing for unavoidable expansions and contractions. The inner cylinder at the head of feed end is connected with the furnace by a hot air flue lined with fire brick. At the discharge end is a revolving head on the inside of which are lifting buckets, so that the material is delivered out through the central casting. The machine has sixteen bearings, thus distributing the load and eliminating danger of heated bearings. The furnace is independent of the machine and located in a convenient place, although generally placed close to the head of the dryer.

The heated air passes through the inner cylinder and returns between the outer and inner cylinders to the fan, passing on the way the material to be dried. By reason of the inclination and revolution of the dryer the material is advanced to the discharge end. This dryer is especially suitable for drying cement rock, clay, coal, ores, sand, gypsum, fullers earth, peat, sewage sludge, tankage, etc., etc.

Class "B" Dryer.—For such materials as oil, kaolin, and ochre, which cannot be dried by direct heat on account of the danger from ignition or injury of the materials by furnace gases, we build a dryer similar in all respects to the Class "A" machine except the gases are taken from the inner flue and returned through a number of tubes.

Class "E" Dryer.—For drying nitrate of soda and other fusible salts which are not injured by direct heat but which cannot be dried in a rotary dryer, on account of the material adhering to the shell, we build a special dryer which has the advantage of direct heat with positive feed and delivery.

Class "F" Dryer.—When the quantity of material to be dried is small or the amount of moisture to be evaporated is light, we build a dryer of single shell construction, and while not as economical in fuel cost as our Class "A" dryer is much lower in first cost.

TINIUS OLSEN AND COMPANY

500 NORTH 12TH STREET, PHILADELPHIA, PA.

TESTING MACHINERY AND INSTRUMENTS

We manufacture a complete line of Olsen testing machines for the Physical testing of any material under any condition. These machines are designed for accuracy, sensitiveness, rigidity, strength and endurance, and they are today the recognized Standard of high grade testing the world over.

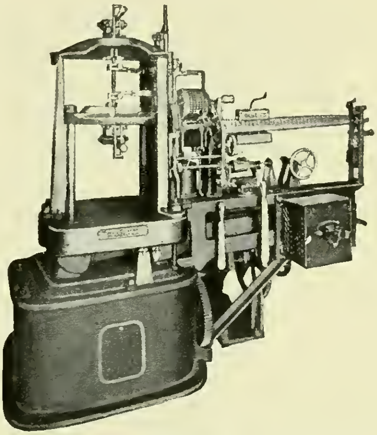
We also are the exclusive manufacturers for the "Turner" impact testing machine, "Fremont" impact testing machine and "White-Souther" endurance testing machine and exclusive American Agents for the "Herbert & Fletcher" file tester and tool steel testing machine.

We build machines for applying all tests as enumerated below and are prepared to design and build special machines for determining any physical property.

Our Catalogue is divided into eight parts as follows:

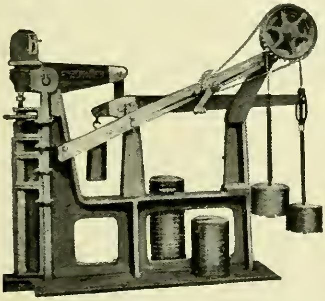
Part A Universal Testing Machines and Instruments.
Part B Spring Testing Apparatus and Spring Machinery.
Part C Cement and Concrete Testing Machinery.
Part D Cloth, Yarn, Paper, Rubber and Leather Testing Machinery.
Part E Wire, Chain and Anchor Testing Machinery.
Part F Oil, Grease, Bearing Metal Testing Machines, Viscosimeters, Dynamometers, etc.
Part G Transverse Testing Machinery.
Part H Special Testing Machinery. (Impact, Indentation, Vibratory, Hardness, Endurance, Bending and Torsion Testing Machinery.)

Any of the above parts mailed upon request.



Olsen's Universal Autographic Testing Machine
Three Screw Type with Direct Motor Drive.
Patented January 7, 1908.

Our Olsen Testing Machines have received the highest awards at the following Expositions:—



Olsen's New Hardness Testing Machine
Patent Applied For.

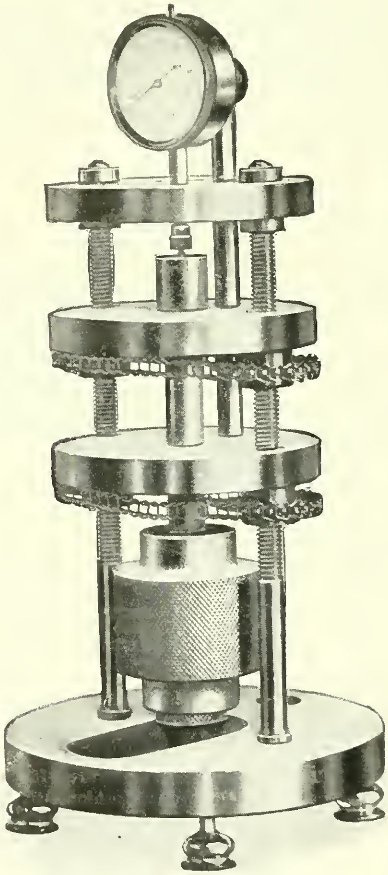
Cincinnati Industrial Exposition, 1881.
International Cotton Exposition, Atlanta, 1881.
World's Industrial & Cotton Centennial Exposition, New Orleans, 1885.
World's Columbian Exposition, 1893.
Export Exposition, Philadelphia, 1899.
Paris Exposition, 1900.
Louisiana Purchase Exposition, 1904.
Jamestown Tercentennial Exposition, 1907.
Alaska-Yukon-Pacific Exposition, 1909.
Also
Elliott-Cresson Gold Medal, Franklin Institute of Philadelphia, Pa.

THE PUSEY & JONES COMPANY

WILMINGTON, DELAWARE, U. S. A.

ENGINEERS AND MANUFACTURERS OF PULP AND PAPER MAKING MACHINERY. MARINE EQUIPMENT. CHEMICAL APPARATUS

THE PLASTOMETER



The Plastometer

The Plastometer measures and indicates the degree of plasticity of rubber, leather, fibre or any other resilient material.

The degree of plasticity is found by measuring the resistance offered by the material to be tested to the penetration of a needle.

Any material may therefore be standardized as to its plasticity.

For example: most of the larger rubber manufacturers have installed Plastometers. In ordering new rubber, or in reordering for duplication, the degree of plasticity may be specified by including in the order the words "Plastometer density of ———." Thus may be insured the duplication of any particular material, while the Plastometer also enables the buyer to check up the plasticity on receipt of the goods.

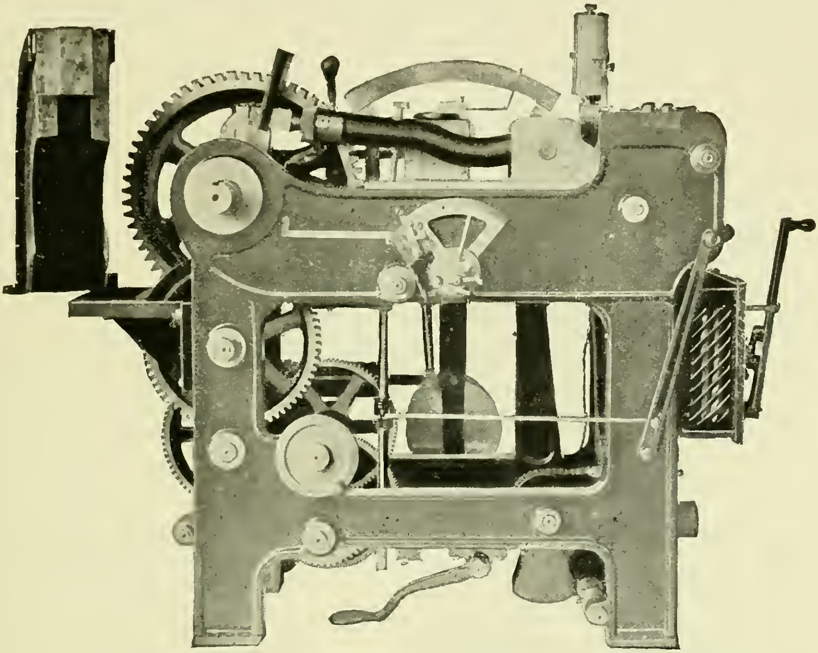
The Plastometer is furnished in a portable mahogany case, and may be easily operated and understood with a few moments' instruction.

The price is $100.00.

SHORE INSTRUMENT & MFG. CO.

555-557 W. 22 ST., NEW YORK, N. Y.

SCIENTIFIC TESTING APPARATUS



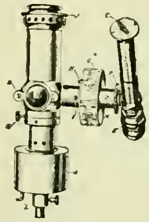
THE SHORE UNIVERSAL COMPENSATING STRENGTH TESTING MACHINE, SET UP FOR CANTILEVER BEND TESTS. (Guard thrown back to show Gear wheel.)

THE SHORE COMPENSATOR (Autographic)

Functions of the Shore Compensator: Takes any size test piece and gives square inch readings on chart. Automatic weighing (therefore fool proof), quick acting and making the following tests:—Cantilever bend test (principal feature), Tension, Gear teeth, Torsion, Compression and shear.

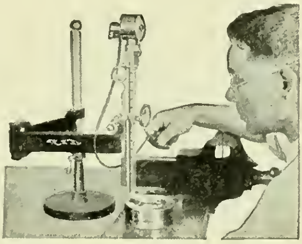
The only machine suitable also for commercial inspection room work. Catalog free.

We make also the *Scleroscope*. The *Pyroscope and Enamelite* (for selective carbonizing).



Pyroscope
Optical Temperature Gauge

Send for Catalogs.

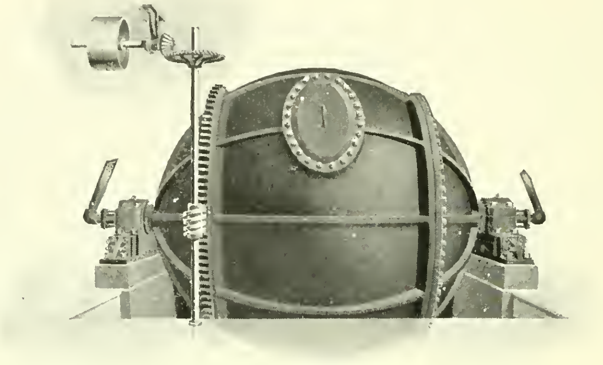


Scleroscope
World Standard Hardness Tester

THE HOOVEN, OWENS, RENTSCHLER COMPANY

HAMILTON, OHIO

SUGAR MACHINERY, CAST IRON GLOBE DIGESTERS AND COOKERS FOR PAPER MILLS AND BLEACHERIES, PLATE GLASS FINISHING MACHINERY, SEMI-STEEL, STEEL RE-INFORCED ANVIL BLOCKS, FLEXIBLE COUPLINGS, AND SPECIAL HEAVY CAST IRON CASTINGS.



Globe Digester and Cooker

The cast iron Globe Digester and Cooker is made in sizes up to and including 12 feet in diameter and is equipped with adjustable babbitted trunnion bearings. The drive may be from the ceiling hangers, as shown in illustration, or from beneath the floor—at the option of purchaser.

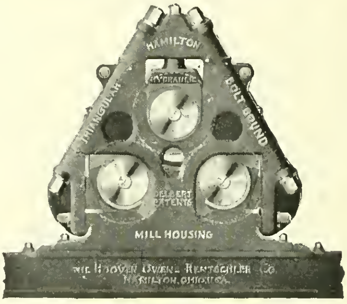
The Anvil Blocks are made in any size and arranged to accommodate any make of hammer. They are composed of a special mixture of semi-steel and thoroughly steel-re-inforced by the HAMILTON process.

The Sugar Mill Housing shown is thoroughly triangular bolt bound, and of such design that the completed housing is of maximum possible strength in every direction. This illustration shows more plainly than any description its excellent design for resisting the enormous pressures to which it is subject.

The 86″ Cylinder illustrated is a good example of our large semi-steel castings and proves that we have excellent facilities for making cast iron or semi-steel castings of any size and weight consistent with modern foundry practice.

Re-Inforced Anvil Block

86″ Semi-Steel Low Pressure Cylinder



Sugar Mill Housing

M. D. KNOWLTON CO.

Main Office and Works

24 ELIZABETH ST., ROCHESTER, N. Y., U.S.A.

NEW YORK LONDON CHICAGO

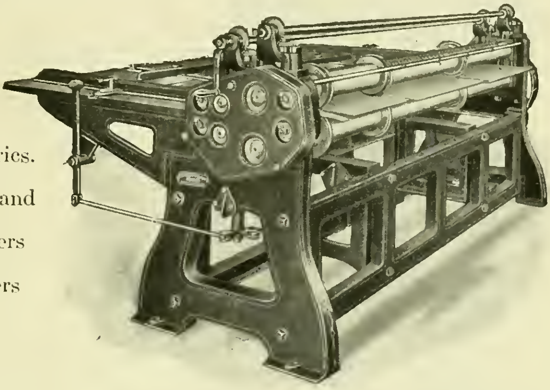
PAPER BOX AND SHIPPING CASE MACHINERY

MACHINES FOR CUTTING, TREATING AND FORMING, PAPER, BOARD AND SHEET FABRICS.

Our experience in designing and making machines for this class of work extends over a period of thirty years. Our shop facilities are excellent. Many of the machines we manufacture are necessary to other industries as well as to that of the paper box. Below is only a partial list of the standard and automatic machines we manufacture and carry in stock. We solicit inquiries for standard and special machinery of this class.

CUTTING MACHINES

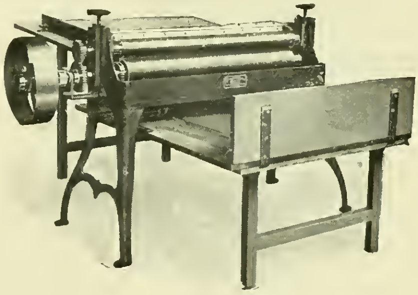
for Slitting, Sheeting or Shape Cutting, Paper, Pulp, Fibre or Corrugated Board and Sheet Fabrics.
Scorers
Rotary Paper Slitters and Rewinders
Rotary Cardboard Slitters
Corner Cutters
Slotters and Flap Cutters
Folding Box Slotters
Roll Sheet Cutters
Neck Choppers
Thumbhole Machines
Skiving Machines
Round and Oval Cutters
Rotary Cutting Knives



Heavy Rotary Slitter for Fibre and Corrugated Board

TREATING MACHINES

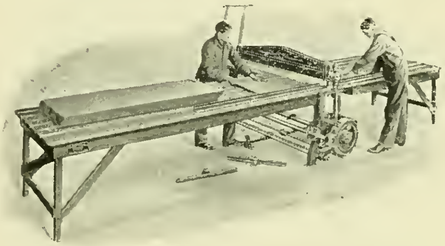
for Coating Paper and Cardboard with Glue, Silicate of Soda, Paste and other Liquid Solutions.
Automatic Gumming Machines
Plain Gumming Machines
Sheet Gluers
Sheet Pasters
Flange Gluers
Folding Box Gluers
Paraffine Coaters
Silicate of Soda Coaters
Glue Cookers and Mixers
Glue Pots



Paraffine Coating Machine

FORMING MACHINES

for Creasing, Bending and Shaping Paper and Cardboard.
Automatic Creasers and Slotters
Folding Box Creasers
Flange Benders
Plain Corner Stayers
Turn-in Corner Stayers
Ender and Corner Stayers
Tape Stayers
Covering Machines
Topping Machines
Trimming Machines



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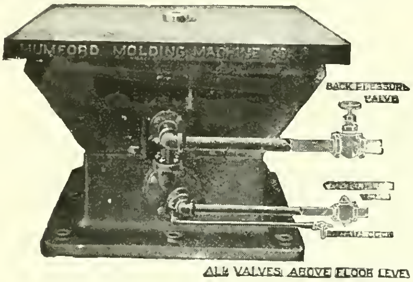

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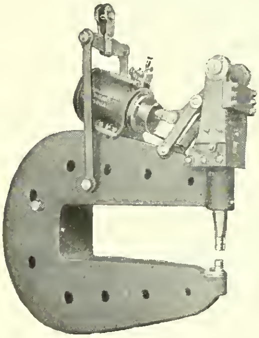

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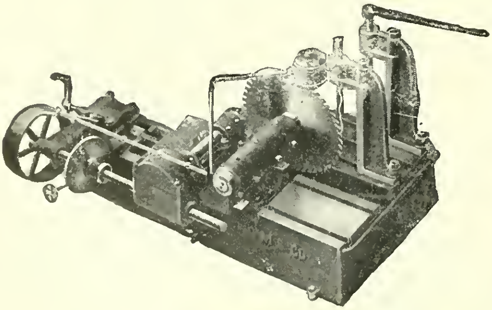

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At infrequent intervals where we have found that for some reason, clearly indicated by us to our inquirers, we were persuaded the conditions would not make the use of our machinery truly profitable to the manufacturers and therefore to us, we do not fail to say so, rather than merit the displeasure of our customers by putting them to needless trouble and probable expense.

We, of course, put before anything else the fulfilling of all guarantees which are clearly stated in our proposals, and strive to do even a little better, so that when a balance is struck between us there remains something to our credit in the good opinion of our customers.

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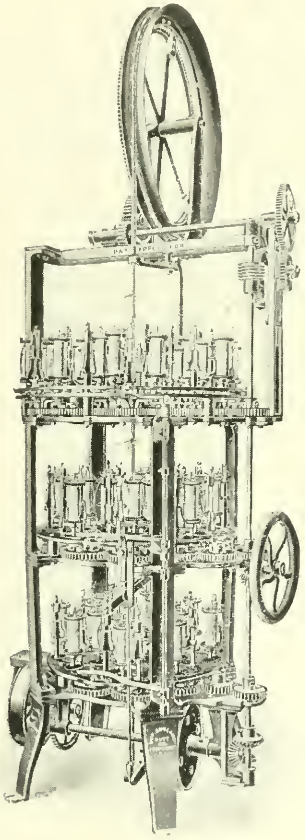

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Rubber Strip Covering Machines, for applying rubber insulation to wires and cables with either single or double seam. These machines are built in several sizes and handle from one up to twenty wires at a time.

NEW ENGLAND BUTT COMPANY

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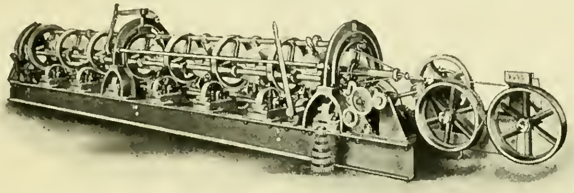
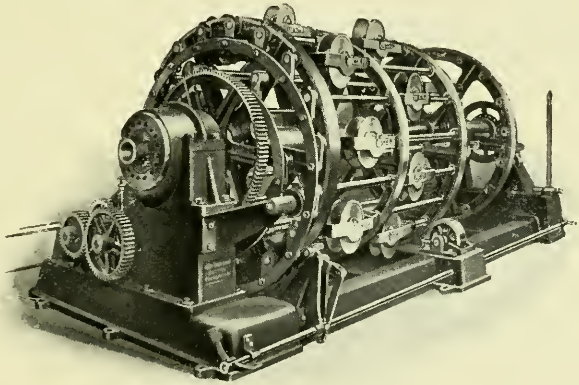


7 REEL HIGH SPEED STRANDING MACHINE.

In this type of machine the reels are carried in stationary cradles hanging at the center of the machine and the frame revolves about them.

By this construction, the machine being perfectly balanced, a very high speed is attainable. It is built in several sizes from small machines, for making the initial strands of small wire ropes, up to large laying machines holding 2000 lbs. to each reel. It is also built for making 19 wire strand.



24 REEL HORIZONTAL CABLING MACHINE.

This machine is of the planetary type in which the reels are kept in a vertical position by means of cranks and an eccentric ring at the rear of the machine.

It is built in single heads or in tandem form with any combination of heads and is used for making cables of concentric strand and also for armoring cables.

We are prepared to furnish this type of machine in a large variety of sizes.

Take up Fixtures for the above machines are built in either the single drum or the double grooved drum types.

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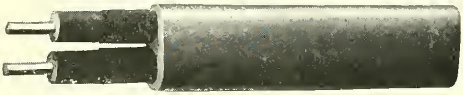

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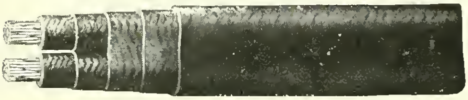

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STRANDED CONDUCTOR VARNISHED CAMBRIC LEAD-ENCASED CABLE
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National Code thickness of rubber as well as heavier thicknesses for special voltages, both ordinary black finish and fireproof finish for station wiring and general use

We manufacture Bare Copper Wires and Cables for Telegraph, Telephone, Street Railway and Long-Distance Power Transmission purposes; also, Insulated Wire of all kinds, such as Weatherproof Wires and Cables for Arc Lighting and Street Railway Feeders, Slow-Burning Wires for Mill Construction; Magnet, Office and Annunciator Wires, Lamp Cord, Submarine Cables, Car Wires together with all kinds of Rubber Wires for the interior wiring of offices and buildings. Our American and Crown Wires are unsurpassed for this purpose. We are also manufacturers of Lead-Encased Cables for underground service, the same being insulated with either rubber, paper or varnished cambric, for electric lighting or power.

The cuts shown herewith illustrate a few of the many varieties of Electrical Wires and Cables made by the American Steel & Wire Company.

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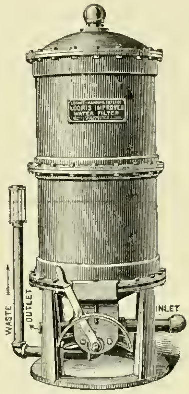


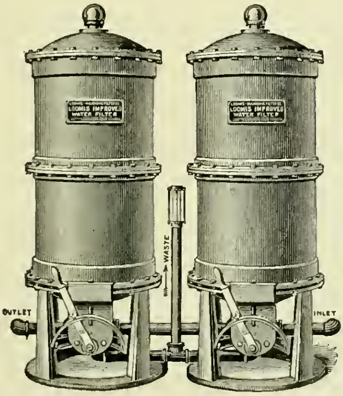
Loomis-Manning Filters are designed with three essential features in view:

Efficiency: This, in a filter, means the ability to thoroughly cleanse the filter bed each time it is washed. For unless the filter bed is kept broken up, free from lumps of all kinds which form around the gelatinous mass which has been collected by the filter bed, the effectiveness of the filter will become less and less until it is nothing. By the reverse flow of the water, the filter bed of a Loomis-Manning Filter is caused to pass up and down through the Loomis Cutting Plate which breaks up the mass of accumulated impurities so that they are purged from the bed and carried off into the waste. The entire bed is in motion, becoming cleansed, at each washing. No part of it is so heavy that it lies stagnant. The bed is a uniform grade of material from top to bottom.

Simplicity: The Manning Single Controlling Valve takes the place of five gate valves and also prevents all chance of errors. By moving one lever to four stations, plainly marked on a dial, the action of the filter may be either filtering, washing the filter bed, filtering to waste (sometimes called re-washing) or the filter may be by-passed. A sight glass is provided so that it is possible to see the quality of the filtered water and watch the cleansing of the filter bed, preventing an extravagant use of water for washing.

Durability: The cylinders and heads are close grained cast iron, to obtain the maximum durability. All materials coming into contact with water are selected to resist corrosive action. Bronze controlling valve, tinned copper screen plates, brass studs, galvanized iron or brass pipe and fittings, etc., reduce repairs to a minimum.

Essential Details: The filter bed rests on tinned copper screen plates, extending under its entire area, bringing about a uniform collection of the filtered water and a uniform distribution of the washing water, insuring a thorough cleansing. The Loomis Confining Plate extends across the filtering chamber near the top and prevents all fish, sticks, stones, etc., from entering and becoming a permanent part of the filter bed. These screen plates are sand tight, preventing any loss of the filtering material into pipe lines, pumps, etc. Baffles are provided in the top and bottom heads to distribute the water as it enters for filtration or for washing. Ample space is provided to enable the bed to wash properly. The coagulant, if necessary, is fed by our indirect method, accurately controlled.



Send for List of Installations.

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THE
JOURNAL
of
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

OCTOBER 1912

35 CENTS A COPY $3 A YEAR

ANNUAL MEETING: NEW YORK, DECEMBER 3-6
MONTHLY MEETINGS: NEW YORK, OCTOBER 8;
BOSTON, OCTOBER 16; CINCINNATI, OCTOBER 24

NEW MEMBERS

The Amendments to the Constitution which are brought to the attention of the membership in this issue of The Journal are of vital importance to the welfare of the Society and should receive the careful consideration of every voting member.

Their adoption, followed by the interest on the part of the entire membership to bring into the Society all prominent engineers not now identified with it, cannot fail to result in a new era of progress. Over 600 members have been admitted during the year, a gain of over 15 per cent.

Coincident with the beginning of the new fiscal year on October 1, the Committee on Increase of Membership opens a new campaign. If each member will give a little attention to this matter *now* there is no reason why a large number of eminent engineers should not be cooperating in the work of the Society this time next year.

Every member is urged to participate in this activity and to invite men of attainment to become identified with the Society. This Committee will give careful attention to all suggestions received, but the success of this undertaking depends upon the interest of the entire membership.

COMMITTEE ON INCREASE OF MEMBERSHIP

I. E. Moultrop, *Chmn.*
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J. V. V. Colwell
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OCTOBER 1912 VOL. 34 No. 10

THE JOURNAL

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

PUBLICATION OFFICE, 29 WEST 39TH STREET . . . NEW YORK

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COMING MEETINGS OF THE SOCIETY

October 5, *Philadelphia, Pa.* Paper, The International Smoke Abatement Exhibition, held in London, March and April 1912, by George H. Perkins.

October 8, *New York City*, Engineering Societies Building, 8.15 p.m. Topic, Smoke Abatement, with report on International Smoke Abatement Exhibition and Conference, held in London, March 1912, by George H. Perkins, head of the Engineering department of the Lowell Textile School, at which he represented the Society and the Lowell Textile School. Discussion by authorities on the subject from other cities.

October 16, *Boston, Mass.* Paper, The Mississippi River Development at Keokuk, Hugh L. Cooper.

October 24, *Cincinnati, Ohio.* Organization meeting of the members in Cincinnati and vicinity, to be addressed by an official representative of the Society.

November 13, *New Haven, Conn.* Annual fall meeting in Mason Laboratory, Sheffield Scientific School.

THE JOURNAL

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOL 34 OCTOBER 1912 NUMBER 10

THE ANNUAL MEETING

Since the close of last winter's season the Annual Meeting, to be held in New York, December 3-6, has been receiving the attention of the Committee on Meetings and sessions have been definitely laid out and arrangements made for the solicitation and examination of papers by sub-committees appointed for that purpose. As already announced, a joint meeting of these sub-committees and of the Committee on Meetings was held in the rooms of the Society last July to discuss plans for the Annual Meeting and to determine which of the committees would be prepared to take a session at that time for the presentation of subjects coming within its scope. There will be sessions held in charge of the following sub-committees: textiles, administration, cement manufacture, machine shop practice, and railroads, and the Gas Power Section will also hold its usual meeting. Other sub-committees also may contribute one or more papers to which parts of sessions will be devoted and there will also be contributions by local sections of papers which have been presented with marked success at monthly meetings in different cities. For so early a date, an unusually large number of papers are already on hand, some of which appear in this and in the September issues of The Journal.

Recently the various committees interested in the entertainment of members and guests at the time of the Annual Meeting have held a conference to discuss ways and means for meeting the expenses of such entertainment. Heretofore these expenses have been met mainly by subscriptions solicited from the New York membership. It is now proposed that certain of the functions, such as the President's Reception which is distinctly a Society affair, shall be paid for out of the general funds, but that it will be in accord with the wishes of the membership at large and with what is coming to be the accepted procedure among the leading organizations, if each member who attends shall bear his proportion of the expenses for the social affairs not strictly related to the work of the Society, such, for example, as the luncheons between sessions and the usual reception and dance on Thursday evening of the Convention. Circulars are now being sent to the membership outlining the plan more in detail and calling for a statement from those who are to attend as to what entertainment will be desired.

Altogether the indications are that the coming Annual Meeting will be among the very best and perhaps the most elaborate which has been held in point of professional meetings and that the entertainment features will be conducted on a better basis.

On Wednesday evening of the meeting there will be a feature in which the entire membership will wish to participate, the reception which the members will tender to Prof. John E. Sweet, Honorary Member and Past-President, to whose work in the formation of the Society all are so greatly indebted.

MEETING IN GERMANY

Arrangements for the meeting of the Society with the Verein deutscher Ingenieure, which promises to be the most remarkable tour of the industries in Germany ever offered to American engineers, are progressing satisfactorily. Mr. Conrad Matschoss, Dozent of the Royal Polytechnic High School of Berlin, is now visiting the United States as a representative of the German society and conferring with the Committee of Arrangements regarding itinerary, professional papers, etc. On the evening of October 8 a dinner will be tendered to Mr. Matschoss by the Council.

In general, the date of departure will probably be June 11, making it possible for the party to arrive in Germany on Saturday, June 21. Here the party will be received by the municipal authorities and given an opportunity to visit the great shipyards, before going on by special train to Leipzig, either on the same or the following day. Hotel reservations in Leipzig have already been made, so that the party will be ensured of satisfactory accommodations. Sunday evening, the 22d, will be the occasion of an informal gathering to promote acquaintanceship, preceding the more formal opening on Monday morning, June 23. At this opening there will be a formal welcome by the King of Saxony and the municipal authorities, and two addresses, one by a representative of the Verein and the other by a representative of our Society, which will probably be on general subjects, such as the history of engineering, the relation of capital and labor, and the effect on industry of technical education. Tuesday, the last day to be spent in Leipzig, will be devoted to technical papers and to various social events and participation in the patriotic gathering in commemoration of the battle of Leipzig, followed by a gala concert by the Gewandhaus Orchestra under Mr. Arthur Nikisch.

The party will leave on Wednesday for a tour of industrial Germany, visiting the principal cities, such as Dresden, Berlin, Essen, where the Baroness von Bohlen (geboren von Krupp) will receive the party at the Krupp works, rarely thrown open to visitors, Dusseldorf and Cologne, with receptions by municipal authorities, and will make a trip up the Rhine as far as Frankfort. At Rudesheim there will be a grand festival on the embankments, and at Mannheim the Hoeschst Chemical Works will be visited. On July 4 the members of the Society will participate in the customary American celebration arranged by the Ambassador to Germany.

The final meeting will be held in Munich about July 10, in connection with the Museum of Technical Arts, of which Dr. von Miller is director. It is expected that the Prince of Bavaria and the Mayor of Munich will also receive the party. The entire trip through Germany and the various official receptions are being arranged by a committee of the German society and there is already the keenest interest shown in the visit by municipal authorities and the heads of industrial establishments.

To assist our Committee of Arrangements it is essential that all those who expect to make the trip shall give their decision at once so that the best accommodations may be secured. The larger the party, the better will be the facilities which the transportation companies will offer. Guests may be included in the trip.

CURRENT AFFAIRS OF THE SOCIETY

The Secretary is able to announce that the subscriptions to the Kelvin Memorial which have thus far been received amount to $510, one hundred and forty members having contributed, and that as soon as complete returns are received the entire fund will be sent to the committee in England, together with the amounts secured by the American Society of Civil Engineers, the American Institute of Mining Engineers, the American Institute of Electrical Engineers, the Canadian Society of Civil Engineers, and the Society of Naval Architects and Marine Engineers. This memorial is a fitting tribute to one of the greatest members of the engineering profession and is one in which every engineer of the English-speaking world should have a part.

The Society history which has been in preparation by the committee appointed by the Council will probably be completed this fall and published in the spring of 1913. The members of the committee are John E. Sweet, H. H. Suplee, and F. R. Hutton, Professor Hutton having been appointed to the place of the late Charles Wallace Hunt.

On August 21, many prominent engineers and business men joined in the congratulations sent to John Fritz, Honorary Member and Past-President of the Society, on his ninetieth birthday. The good wishes of the Society were conveyed to Mr. Fritz by the Secretary in a telegram.

Calvin W. Rice, *Secretary.*

CORRECTION IN SEPTEMBER ISSUE OF THE JOURNAL

Attention is drawn to the following error, occurring on page 1370, line 11, of the September issue of The Journal, in the closure of A. G. Christie: " the efficiency ratio of the unit shall not exceed that represented by the guarantee, " should read, " the efficiency ratio of the unit shall not *be less than* that represented by the guarantee."

NEW HAVEN MEETING

The fall meeting of the New Haven Section will be held on Wednesday, November 13, with afternoon and evening sessions, at the Mason Laboratory of the Sheffield Scientific School. In the announcement of this meeting issued by the Committee on Meetings for New Haven, the following four subjects are suggested for future meetings upon which members in the vicinity are asked to express their choice and to offer papers for discussion: the market for Connecticut manufactured products; garbage disposal for cities; the transportation of materials in manufacturing plants; and education for engineers.

REPORT OF THE NOMINATING COMMITTEE

The Secretary, according to B 29 of the Constitution and By-Laws, announces the receipt of the following report from the Nominating Committee, consisting of Ambrose Swasey, Chairman, A. E. Cluett, W. B. Gregory, John Hunter, William Schwanhausser: For President, W. F. M. Goss, Urbana, Ill.; Vice-Presidents (for two years), James Hartness, Springfield, Vt., I. E. Moultrop, Boston, Mass., H. G. Stott, New York; Managers (for three years), W. B. Jackson, Chicago, Ill., H. M. Leland, Detroit, Mich., Alfred Noble, New York; Treasurer, Wm. H. Wiley. All of the nominees have been notified and have accepted.

TWO INTERNATIONAL CONGRESSES

During the past month two international congresses have been held in this country: the Sixth Congress of the International Association for Testing Materials and the Eighth International Congress of Applied Chemistry.

THE INTERNATIONAL ASSOCIATION FOR TESTING MATERIALS

The International Association for Testing Materials met in the Engineering Societies Building, New York, September 2 to 7, with an attendance of more than 700 delegates from 28 different countries. On the evening of the opening day a reception was tendered the delegates by the Society, in association with the American Society of Testing Materials and the founder societies.

The movement which resulted in the establishment of this association was started in 1882 by J. Bauschinger, for the purpose of establishing uniform methods of testing the most important materials of construction. Four conferences were held under the chairmanship of the founder: at Munich in 1884, at Dresden in 1886, at Berlin in 1890 and at Vienna in 1893. At first only nations speaking the German language were represented, but the conferences rapidly assumed an international character. In 1893 the United States was represented by Gustavus C. Henning, Mem.Am.Soc.M.E.

The first congress met in Zurich in 1895, L. Tetmajer, president, and it was here decided to replace the somewhat informal meetings by a permanent organization under the name of the International Association for Testing Materials. In the same year was started the publication of the official organ of the association, Die Baumaterialienkunde. The second congress took place in Stockholm in 1897 when the association was joined by English members of the Iron and Steel Institute. Such papers as Osmond's Metallography as a Method of Testing, Rejto's Internal Friction, and Carnot's Chemical Analysis of Iron, presented before that meeting have a definitely recognized place in the history of engineering. In 1901 when the association held its third congress in Budapest, the membership had grown to 1800. Two papers made this meeting a notable one, Considère's Properties of Reinforced Concrete, which was then only slightly known as a material of construction, and Baron Jornstorff's, which discussed the forms of carbon in iron and showed that the chemical analysis of metals as carried on up to that time was absolutely inadequate, and that a rational method of analysis was required to determine the forms and combinations of the constitutent elements of metals.

For the fourth congress in Brussels in 1906 Le Chatelier and Guillet organized a laboratory containing some of the most modern testing machinery for both mechanical and metallographic tests of metals. Many of the papers presented before the meeting were the results of the work of international commissions. This is an important feature of the association for it both insures a high quality of the papers submitted and facilitates the adoption of the resolutions of the congresses by official bodies in the various countries.

The fifth congress was held in Copenhagen, with 900 members present, representing 22 countries. Particular attention was paid to metallography: Professor Schule, Dr. Leon and Dr. Ludwik discussed hardness and impact testing, while Dr. Moldenke, Mem.Am.Soc.M.E., presented the official report on uniform methods of testing cast iron. Valuable papers were also read on the testing and properties of cement, stone and concrete. An important branch of the work of the association is the establishment and adoption of international specifications and nomenclature, the report of the committee of the uniform nomenclature of iron and steel, especially the nomenclature of the microscopic constituents of iron and steel, by H. M. Howe and A. Sauveur of this country, being an example.

The association now numbers about 2800 members representing about every country of the world. Among the papers presented at the sixth congress may be mentioned the following: C. Grard (Studies of Hardness of Steels) showed that hardness tests may be used in place of tension tests by multiplying the hardness number by a constant, or may serve for tests of homogeneity, and specified the precautions necessary to attain the required precision; the author considered it practicable to specify hardness and impact test requirements in steel specifications. G. Charpy (committee paper on Impact Tests) stated that the tests made by the committee with the two sizes of bars approved by the fifth congress (30 by 30 mm., 120 mm. span, and 10 by 10 mm., 40 mm. span) proved that equality of results is not bound to hold, and that smaller bars give smaller values. M. Derihon (Remarks on the Brittleness Test) believed, after having used the Fremont impact test, that was a reliable means of discovering brittleness and was not an unduly severe method of test; the author had used it since 1904, with up to 12,000 tests per month, for all steel bought or treated in his plant, and now had only 0.3 per cent rejections. C. Fremont (A New Testing Method for Steel Tubes) suggested application of the notched bar impact test to tubes and said if a narrow ring of the tube were flattened, doubled to U-shape, notched by a saw cut, and tested in the Fremont machine it should give at least 16 kgm. per sq. cm. (1072 ft-lb. per sq. in.) initial sectional area.

T. E. Stanton gave an account of the recent researches at the National Physical Laboratory at Teddington on the resistance of metals to alternating stresses. He said that rapidity of alterations had no influence on the results up to 7000 per minute, and maximum range of stress was about the same as the high yield-point value; strength for long-time endurance was apt to be in the reverse order of the single blow impact strengths, while repeated impact strength could be computed from the simple fatigue strength. O. Boudouard (Endurance Test of Metals) presented a new method of testing by measuring the damping of vibrations. C. W. Burrows showed that there was a close relationship between the magnetic properties and the presence of flaws, blowholes and strains, while elastic limit, tensile strength, and hardness were, within certain limitations, functions of the magnetic reluctance and electric resistance of the materials. He believed therefore that certain electric and magnetic tests may be used to supplement or replace certain mechanical tests. Several other papers indicated the existence of the same relationship between the mechanical and magnetic properties of steel. F. Robin in his paper, Several Mechanical Properties of Metals in the Hot State, demonstrated that iron and pearlitic steel under static stress showed a minimum strength at about 100 deg., a maximum at about 250 deg., and a bend in the curve at about 500 deg. Under dynamic stress the minimum was found near 300 deg. and the maximum at about 500 deg. cent. The effect of the coldworking of hard metals grew with the temperature and tended to increase their resistant qualities; on the other hand, heating diminished the resistance progressively, and the two effects would bring about a minimum of the properties of resistance, which accidentally fell near 100 deg. cent. O. Gallander showed how the depth of a crack in a conductive metal may be determined electrically, while A. Mesnager (A Method of Foreseeing Rail Fractures) claimed that surface fissures on the head of a rail have a great influence in causing fractures, and a picture of the surface of a rail with its fissures and defects may help to determine the primary cause of an accident in a case of derailment, by showing whether or not the rail was brittle on account of internal fissures. W. Campbell and J. Glassford found, from a study of 23 specimens of pig, cast iron and steel subjected to prolonged exposure to superheated steam or many repetitions of heating and cooling, that white cast-iron, malleable castings and steel, oxidized superficially, while in gray cast iron the oxidation penetrated deeper, following the graphite flakes, the change increasing with the silicon content. C. Fremont advocated shearing as a method of mechanical testing of cast iron. He proposed the use of a test-piece 5 mm. square worked from a 7.5 mm. cylinder taken out of the casting with a core drill; shearing this test-piece every 3 mm. along its length gave data on strength at different depths below the surface. T. D. West, Mem.Am.Soc.M.E., (A Suggested System of Test-bars for Chillable Irons) stated that in testing chillable irons both the white and the gray phase should be tested, and described the molds for both. R. Moldenke, Mem.Am.Soc.M.E., criticized the tendency to push unduly the quality of a cheap material to its upper limit, rather than to step at once to the higher grade material. De Nolly and Veyret after studying the gases disengaged by various carburizers to determine the cause of explosions which occur in case hardening, concluded that hydrogen emitted at higher temperatures by the case-hardening materials was the probable cause of these explosions. They recommended therefore that carburizers containing much animal or vegetable matter not sufficiently calcined be not used, but instead well calcined charcoal alone or mixed with barium carbonate, the heating being slow up to 700 deg. cent. (1292 deg. fahr.). N. Pecoraro (Researches on White-metals) stated that the temperature of bearing metals seemed to have a distinct influence on their resistance to wear, and the only mechanical tests having practical value are the hardness test, impact tests which may give indications as to brittleness, and such tests in gradual compression as measure the value of the metal for supporting its working loads. C. Fremont described a new testing method for steel wire, being an adaptation of the impact tests with the wire notched with a file.

In section B, Cement and Stone, several papers were devoted to the determination of the very fine powder in portland cement, but without any definite proposals having been made as yet by the committee. E. G. Coker, by applying the method of polarized light effects in transparent bodies under stress to celluloid models of cement of the English, American, and continental forms established the fact that different test results would be shown in cement tests with briquettes of different forms. J. Y. Jewett in describing the tests of construction materials by the United States Reclamation Service called attention, among other things, to the superiority of sands made from crushed rock, such as granite, limestone and basalt, over natural quartz sands, for the production of concrete, while Bertram Blount (Accelerated Tests for Constancy of Volume of Cement) stated that no case had been found where a cement which stood the Le Chatelier test proved unsound later, and that since this test had been introduced in England there had been no difficulty in obtaining cement that would stand it. Speaking on the basis of experiments made by the Association of German Portland Cement Works, M. Gary stated that comparative tests showed that the Le Chatelier test did not give concordant results in different laboratories, and that the material and make of the apparatus influenced the results; no cement proved unsound in use after passing the standard cold-water test, but the boiling test rejected many cements which proved wholly sound in use. The last statement was not accepted by F. Schule who believed that cements which are to be used in dry exposure should be subjected to the boiling test, as is done in Switzerland.

The association continued its important work on the action of seawater on concrete. V. J. P. de Blocq van Kuffeler reported that experience in Holland indicated that reinforced concrete was sufficiently permanent to warrant its use for marine construction, but the following rules should be strictly adhered to: only slow-setting portland cement should be used; mixing and ramming to be done with great care; trass or other puzzolanic additions are desirable; the mortar should be dense and the mixture rich; setting in the moist air before immersion in the sea is advantageous. The Porosity of Mortars was the subject of a paper by J. Bied who found that an important degree of impermeability to diffusion could be imparted to mortars exposed to the action of seawater by adding to them at the time of mixing barium aluminate transformed into barium sulphate, the latter, in precipitating, closing the capillary pores; a certain degree of impermeability could also be secured by the addition of fatty or gelatinous materials, such as mineral oils.

In Section C, Miscellaneous Materials, there were presented several papers on the testing of paint materials, treating, among other things, of the influence of the pigment on the underlying metal (A. S. Cushman: Paints for Metallic Structures). G. Janka proposed the inclusion of hardness pressure of wood by the ball-pressure method among the standard timber tests of the association; the hardness of wood was found to correspond generally to the crushing strength, with the exception, however, of trees of eccentric trunk growth where the wide-ringed side was harder than the other, but lower in strength. L. Fea discussed the Practical Determination of the Viscosity of Oils, stating that the Engler figures can be taken as equivalent to absolute measures (it was found that the Engler relative viscosities were linear functions of the Pagliani absolute viscosities which were alike for all the oils), and the wear coefficient could be taken as a linear function of viscosity.

A considerable lack of agreement was evidenced with respect to the adoption of international specifications for materials of construction, both in the section of metals (iron and steel), and in section B, with respect to the standardization of the content of SO_3 in portland cement. Some delegates objected to the adoption of any such specifications partly on principle, because they considered that such an act would involve the International Association in matters lying outside of the domain of the science of engineering; others objected to such specifications, particularly those with regard to portland cement, because their adoption might place at a disadvantage their home industries. As a result, in the case of iron and steel specifications the committee was transformed from a legislative body into one whose work will be the compilation and dissemination of existing specifications in different countries, while the committee on the content of SO_3 in portland cement was simply instructed to collect more material.

Unlike the former congresses of the association, there were no papers defending the status of metallography of iron and steel. The new science has evidently obtained a recognized position among its older peers, and its status cannot be questioned any more.

Another interesting feature of the congress was the comparatively large number of papers devoted to the testing of materials while in actual use, either by electric or magnetic measurements, or by the investigation, in case of rails, of outside fissures. President Howe in his opening address went so far as to ask, "Who shall say that the pitch of volume or timbre of sound emitted by a rail as a result of a given excitement may not be made to disclose pitilessly its hidden defects and to measure the fitness, not alone of the material of which it is composed, but of the rail as a whole structure? Or, giving rein to our fancy, we hear the inspector report, 'This 100-story building indeed responds to G-sharp, but its timbre has this abnormality and these harmonics are exaggerated.'" The sixth congress was practically the first to bring forward as a practical proposition the testing of metals by indestructible methods.

The next meeting is to be held in St. Petersburg in 1915, with Professor N. Belelubsky, of St. Petersburg, as President.

INTERNATIONAL CONGRESS OF APPLIED CHEMISTRY

The eighth International Congress of Applied Chemistry held in Washington and New York, August 31 to September 11, was attended by about 350 foreign delegates and a total of 2223 persons. The bulk of the papers presented was printed in advance of the meetings and represented the tremendous expenditure of labor incident to the preparation of 24 finely printed volumes. Owing to the difficulty of securing a sufficient number of stenographers for reporting the proceedings in their several languages, an ingenious scheme of reporting the discussions by phonograph was adopted, which, however, did not prove to be an unqualified success.

Although primarily a chemical congress, it contained many papers of considerable interest to the mechanical engineer as well. C. G. Fink reported on the applications of ductile tungsten, a remarkable material, with the melting point higher than that of platinum, tensile strength higher than that of iron and nickel, practically insoluble in all of the common acids, and so hard that it will readily scratch glass and still be ductile. Another comparatively new metal, manganese bronze, and methods of its testing was the subject of a paper by Wm. M. Corse and V. Skillman who laid particular emphasis on the selection of proper test coupons, and using methods of testing corresponding to the type of coupon selected. The electric furnace naturally attracted much attention. D. A. Lyon pointed out that it was not brought out with the idea of competing with the blast furnace, but for the purpose of developing a furnace and a process which could be utilized where high cost prevents the use of the blast furnace. The commercial operation of the electric furnace depends upon the development of cheap electric power, but outside of water power even the gas engine does not appear at present to be cheap enough to permit of its use for iron reduction furnace work. Bradley Stoughton discussed the still mooted question of the influence of titanium on the strength of iron castings. Only 0.9 lb. per ton of steel is required to give increased resistance to wear, but the treatment must be very carefully carried out if good results are to be obtained.

In the section on cement W. A. Schmidt presented a paper on the Control of Dust in Portland Cement Manufacture by the Cottrell Prescription Process, i.e. by electrostatic means, while W. C. Reibling and F. D. Reyes discussed the physical and chemical properties of portland cement and made recommendations for increasing the efficiency of the present standard cement specifications of the American Society for Testing Materials.

J. C. Olsen and A. E. Ratner presented the results of their experiments on the decomposition of linseed oil during drying, showing that a considerable amount of gases is given off during that process (carbon and hydrogen). Several papers were presented on lubricating oils: T. T. Gray gave the results of comparative tests made with the Engler-Saybolt instruments, with the ratios of the readings of the two instruments given in diagrams, so as to permit a ready conversion from Engler's specific viscosities to Saybolt viscosities and vice versa. H. C. Porter and F. K. Ovitz discussed the question of spontaneous combustion of coal: temperature of the surrounding space is a large factor, and storing of coal in warm places is greatly conducive to spontaneous combustion; the presence of dust and fine coal is also to be avoided as much as possible. The rate of oxidization of coal is a function of the partial pressure of oxygen present, and therefore of the rate of flow of air, but freshly mined coal absorbs oxygen more readily than coal which has been exposed to the air some time.

The following two statements made by Dr. Karl Duisberg, of Germany, are particularly interesting because of the authority of the speaker. The first refers to the production of cold:

> It is hoped in the near future to produce refrigerating machines which, by the use of suitable hydrocarbons, will give temperatures of - 80 deg. cent. Plants for the liquefaction of air, producing as low a temperature as - 190 deg. cent., are becoming more and more common, and are especially profitable where gas mixtures rich in oxygen, or where pure nitrogen simultaneously produced, can be utilized. The Badische Anilin and Soda Fabrik, in Ludwigshafen on the Rhine, intend to manufacture hydrogen from water gas in a similar way and to utilize the carbon monoxide, which is simultaneously obtained, as a source of power.

Another important statement refers to the commercial production of artificial rubber:

> The difficulties which have been overcome were great indeed and those which still remain to be surmounted, in order to produce a substance equal to para caoutchouc in quality and capable of competing with cheap plantation rubber costing only 2 marks per kilo, are still greater. The end in view is that artificial rubber may play as important a role in the markets of the world as does natural rubber. I do not know when synthetic rubber will bring the millions which prophets see in its exploitation, surely not in the immediate future, although synthetic rubber will certainly appear on the market in a very short time. But I hope to live long enough to see Art triumph here also over Nature.

The next Congress of Applied Chemistry will be held in St. Petersburg, Russia, in 1915, Professor Walden, of the Riga Polytechnicum, President.

AMENDMENTS TO CONSTITUTION

RESPECTING ELECTION OF MEMBERS

At the Spring Meeting, Mr. Jesse M. Smith, at the request of the Council, offered the amendments suggested by the Committees on Membership and Increase of Membership.

In general they make the grades of membership similar to the more important American and English societies and simplify the method of election and make it also similar to most societies.

This is accomplished by adding the engineering grade of Associate-Member, intermediate between Member and Junior, and by substituting election by the Council for election by the membership.

Wherever the grade of Associate-Member should be added and the method of electing members should be changed in existing paragraphs of the Constitution, it appears in the subjoined.

The following are the proposed amendments which will be open for discussion at the Annual Meeting in December:

AMENDMENTS TO THE CONSTITUTION TO BE PRESENTED UNDER PROVISION C57

AT THE

ANNUAL MEETING IN NEW YORK, DECEMBER 4, 1912

Present	*Proposed*
C6 The membership of the Society shall consist of Honorary Members, Members, Associates, and Juniors. Honorary Members, Members, and Associates are entitled to vote and to hold office. Juniors shall not be entitled to vote nor to be officers of the Society, but shall be entitled to the other privileges of membership.	C6 The membership of the Society shall consist of Honorary Members, Members, Associates, Associate-Members and Juniors. Honorary Members, Members, Associates and Associate-Members are entitled to vote and to hold office. Juniors are not entitled to vote nor to hold office but are entitled to all the other privileges of membership.
C7 Honorary Members, Members and Associates are entitled to vote on all questions before any meeting of the Society, in person or by proxy, given to a voting member. A proxy shall not be valid for a greater time than for six months.	C7 Honorary-Members, Members, Associates and Associate-Members are entitled to vote on all questions before any meeting of the Society, in person, or by proxy given to a person entitled to vote. A proxy shall not be valid for a greater time than for six months.

C8 Honorary Members shall be persons of acknowledged professional eminence, and their number shall not exceed 25 at any time.	C8 An Honorary-Member shall be a person of acknowledged professional eminence. The number of Honorary-Members shall not exceed 25 at any time.
C9 A member shall be thirty years of age or over. He must have been so connected with Engineering as to be competent, as a designer or as a constructor, to take responsible charge of work in his branch of engineering, or he must have served as a teacher of engineering for more than five years.	C9 A Member shall be an Engineer or Teacher of Applied Science of thirty years of age or over. He must have shown by his experience or by the responsibility of his duties that he is competent to take charge of work in his profession.
C10 An Associate shall be thirty years of age or over. He must have been so connected with some branch of engineering or science, or the arts, or industries, that the Council will consider him qualified to coöperate with engineers in the advancement of professional knowledge.	C10 An Associate shall be thirty years of age or over. He need not be an Engineer but must have been so connected with some branch of Engineering or science, or the arts, or industries, that the Council will consider him qualified to coöperate with Engineers in the advancement of professional knowledge.
	New Article—*insert between* C10 *and* C11 C— An Associate-Member shall be an Engineer or Teacher of Applied Science of twenty-five years of age or over. He must show by his experience or by his duties, that he is competent to execute work in his profession.
C11 A Junior shall be twenty-one years of age or over. He must have had such engineering experience as will enable him to fill a responsible subordinate position in engineering work, or he must be a graduate of an engineering school. A person who is over thirty years of age shall not be eligible to membership in the Society as a Junior.	C11 A Junior shall be twenty-one years of age or over. He must have had such engineering experience as will enable him to fill a subordinate position in engineering work, or he must be a graduate of an engineering school.
C12 The rights and privileges of every Honorary Member, Member, Associate, and Junior shall be personal to himself, and shall not be transferable or transmissible by his own act or by operation of law.	C12 The rights and privileges of every Honorary-Member, Member, Associate, Associate-Member and Junior shall be personal to himself, and shall not be transferable or transmissible by his own act or by operation of law.
C14 All applications for membership to the grades of Member, Associate, or Junior, shall be presented to the Council, which shall consider and act upon each application, assigning each approved applicant to the grade of membership to which, in the judgment of the Council, his qualifications entitle him. The name of each candidate thus approved by the Council shall, unless objection is made by the applicant, be submitted to the voting membership for election, by means of a letter-ballot.	C14 All applications for membership to the grades of Member, Associate, Associate-Member or Junior, shall be presented to the Council, which shall consider and act upon each application, assigning each approved applicant to the grade of membership to which, in the judgment of the Council, his qualifications entitle him. The name of each candidate thus approved by the Council shall, unless objection is made by the applicant, be submitted to all the members of the Council for election, by means of a letter-ballot.
C15 Associates or Juniors desiring to change their grade of membership shall make application to the Council in the same manner as is required in the case of a new applicant.	C15 Any person desiring to change his grade of membership shall make application to the Council in the same manner as is required in the case of a new applicant.
C16 Election to membership shall be by sealed letter-ballot as the By-Laws shall provide. Adverse votes to the number of two per cent of the votes cast shall be required to defeat the election of an applicant for any grade of	C16 Election to membership shall be by sealed letter-ballot of the Council as the By-Laws shall provide. Fifteen affirmative votes shall be required for the election of a candidate to membership in any grade. One negative

C16—*Continued*

membership. The Council may, in its discretion, order a second ballot upon a defeated applicant, in which case adverse votes to the number of four per cent of the votes cast shall be required to defeat the election.	vote shall defeat an election to Honorary Membership. Two negative votes shall defeat an election to any other grade.
C17 The Election of Honorary Members shall be by vote of the Council taken by letter-ballot, as provided in the By-Laws. One dissenting vote shall defeat such election.	C17 Strike out altogether present C17.
C19 The initiation fee for membership in each grade shall be as follows: For Member....................$25.00 For Associate.................... 25.00 For Junior...................... 15.00	C19 The initiation fee for membership in each grade shall be as follows: For Member....................$25.00 For Associate.................... 25.00 For Associate-Member........... 25.00 For Junior...................... 15.00
C21 The annual dues for membership in each grade shall be as follows: For Member....................$15.00 For Associate.................... 15.00 For Junior...................... 10.00 for the first six years of his membership and thereafter the same as for an Associate.	C21 The annual dues for membership in each grade shall be as follows: For Member....................$15.00 For Associate .... 15.00 For Associate-Member........... 15.00 For Junior...................... 10.00 for the first six years of his membership and thereafter the same as for an Associate-Member.
C22 The Council may, in its discretion, permit any Member or Associate to become a Life Member in the same grade, by the payment at one time of an amount sufficient to purchase from the Equitable Life Assurance Society of New York, an annuity on the life of a person of the age of the applicant equal to the annual dues in his grade. Such Life Member shall not be liable thereafter for annual dues.	C22 The Council may, in its discretion, permit any Member, Associate or Associate-Member to become a Life-Member in the same grade, by the payment at one time of an amount sufficient to purchase from the Equitable Life Assurance Society of New York, an annuity on the life of a person of the age of the applicant equal to the annual dues in his grade. Such Life-Member shall not be liable thereafter for annual dues.
C24 Any Member, Associate, or Junior, who shall leave his annual dues unpaid for one year, shall not receive the volume of Transactions until such arrears are paid. Any Member, Associate, or Junior, who shall leave his dues unpaid for two years, shall, in the discretion of the Council, have his name stricken from the roll of membership, and shall cease to have any further rights as such.	C24 A Person in any grade of membership who shall leave his annual dues unpaid for one year, shall not receive the volume of Transactions until such arrears are paid. A person who shall leave his dues unpaid for two years, shall, in the discretion of the Council, have his name stricken from the roll of membership, and shall cease to have any further rights as such.
C26 The affairs of the Society shall be managed by a Board of Directors chosen from among its Members and Associates, which shall be styled "The Council." The Council shall consist of the President of the Society, who shall be the presiding officer, six Vice-Presidents, nine Managers, the Treasurer, and five Past-Presidents. Five members of the Council shall constitute a quorum for the transaction of business. The Secretary may take part in the deliberations of the Council, but shall not have a vote therein. The Chairman of the Finance Committee shall attend the meetings of the Council and take part in the	C26 The affairs of the Society shall be managed by a board of directors chosen from among its Members, Associates and Associate Members which shall be styled "The Council." The Council shall consist of the President of the Society, who shall be the presiding officer, six Vice-Presidents, nine Managers, the Treasurer, and the five surviving Past-Presidents who have last held office. Five members of the Council shall constitute a quorum for the transaction of business. The Secretary may take part in the deliberations of the Council, but shall not have a vote therein. The Chairman of the Finance Committee shall attend the meetings of the

C26—*Continued*

discussion of financial questions, but shall not have a vote.	Council and take part in the discussion of financial questions, but shall not have a vote.
C27 The five surviving Past-Presidents who last held the office shall be the members of the Council, with all the rights, privileges, and duties of the other members of the Council.	C27 Strike out present C27.
C29 Should a vacancy occur in the Council, or in any elective office except the Presidency, through death, resignation or other cause, the Council may elect a Member or Associate to fill the vacancy until the next annual election.	C29 Should a vacancy occur in the Council, or in any elective office except the Presidency, through death, resignation or other cause, the Council may elect a Member, Associate or Associate-Member to fill the vacancy until the next annual election.
C33 The Council may, by a two-thirds vote of the members present, declare any elective office vacant, on the failure of its incumbent for one year, from inability or otherwise, to attend the Council meetings, or to perform the duties of his office, and shall thereupon appoint a Member or Associate to fill the vacancy until the next Annual Meeting. The said appointment shall not render the appointee ineligible to election to any office.	C33 The Council may, by a two-thirds vote of the members present, declare any elective office vacant, on the failure of its incumbent for one year, from inability or otherwise, to attend the Council meetings, or to perform the duties of his office, and shall thereupon appoint a Member, Associate or Associate-Member to fill the vacancy until the next Annual Meeting. The said appointment shall not render the appointee ineligible to election to any office.
C34 At each Annual Meeting there shall be elected from among the Members and Associates: A President, to hold office for one year. Three Vice-Presidents, each to hold office for two years. Three Managers, each to hold office for three years. A Treasurer, to hold office for one year.	C34 At each Annual Meeting there shall be elected from among the Members, Associates and Associate-Members: A President, to hold office for one year. Three Vice-Presidents, each to hold office for two years. Three Managers, each to hold office for three years. A Treasurer, to hold office for one year.
C41 The Society shall hold two meetings in each year. The Annual Meeting shall begin in New York City on the first Tuesday in December, and a Semi-Annual Meeting shall be held at such time and place as the Council may appoint. Fifty Members and Associates shall constitute a quorum for the transaction of business.	C41 The Society shall hold two meetings in each year. The Annual Meeting shall begin in New York City on the first Tuesday in December, and a Semi-Annual Meeting shall be held at such time and place as the Council may appoint. Fifty persons entitled to vote shall constitute a quorum for the transaction of business.
C42 Special Meetings of the Society may be called at any time at the discretion of the Council, or shall be called by the President upon the written request of fifty members entitled to vote, the notices for such meetings to state the business for which such meeting is called and no other business shall be entertained or transacted at that meeting.	C42 Special meetings of the Society may be called at any time at the discretion of the Council, or shall be called by the President upon the written request of fifty persons entitled to vote, the notices for such meetings to state the business for which such meeting is called and no other business shall be entertained or transacted at that meeting.

Amend C45, by adding at the end of the list of Standing Committees the words: "Committee on Constitution and By-Laws."

C45 The Standing Committees of the Society to be appointed by the President shall be:

Finance Committee.
Committee on Meetings.
Publication Committee.
Membership Committee.
Library Committee.
House Committee.
Research Committee.
Public Relations Committee.
Committee on Constitution and By-Laws.

Present	*Proposed*
C48 Special Nominating Committee: Twenty or more members entitled to vote may constitute themselves a Special Nominating Committee, with the same powers as the Annual Nominating Committee.	C48 Special Nominating Committee: Twenty or more persons entitled to vote may constitute themselves a Special Nominating Committee, with the same powers as the Annual Nominating Committee.
C57 At any semi-annual meeting of the Society any member may propose in writing an amendment to this Constitution. Such proposed amendment shall not be voted at that meeting, but shall be open to discussion and to such modification as may be accepted by the proposer. The proposed amendment shall be mailed in printed form by the Secretary to each member of the Society entitled to vote, at least sixty days previous to the next annual meeting, accompanied by comment by the Council, if it so elects. At that annual meeting such proposed amendment shall be presented for discussion and final amendment, and shall subsequently be submitted to all members entitled to vote, provided that twenty votes are cast in favor of such submission. The final vote on adoption shall be by sealed letter-ballot, closing at 12 o'clock noon, on the first Monday of March following.	C57 At any semi-annual meeting of the Society any person entitled to vote may propose in writing an amendment to this Constitution. Such proposed amendment shall not be voted at that meeting, but shall be open to discussion and to such modification as may be accepted by the proposer. The proposed amendment shall be mailed in printed form by the Secretary to each person entitled to vote, at least sixty days previous to the next annual meeting, accompanied by comment by the Council, if it so elects. At that annual meeting such proposed amendment shall be presented for discussion and final amendment, and shall subsequently be submitted to all persons entitled to vote, provided that twenty votes are cast in favor of such submission. The final vote on adoption shall be by sealed letter-ballot, closing at 12 o'clock noon, on the first Monday of March following.
C58 The letter-ballot, accompanied by the text of the proposed amendment, shall be mailed by the Secretary to each member of the Society entitled to vote, at least thirty days previous to the closure of the voting. The ballots shall be voted, canvassed, and announced as provided in the By-Laws. The adoption of the amendment shall be decided by a majority of the votes cast. An amendment shall take effect on the announcement of its adoption by the Presiding Officer of the semi-annual meeting next following the closure of the vote.	C58 The letter-ballot, accompanied by the text of the proposed amendment, shall be mailed by the Secretary to each person entitled to vote, at least thirty days previous to the closure of the voting. The ballots shall be voted, canvassed, and announced as provided in the By-Laws. The adoption of the amendment shall be decided by a majority of the votes cast. An amendment shall take effect on the announcement of its adoption by the Presiding Officer of the semi-annual meeting next following the closure of the vote.

Renumbering

Renumber Articles of the Constitution after they have been approved by the Society.

THE BALTIMORE SEWERAGE PUMP VALVE

By A. F. Nagle

ABSTRACT OF PAPER

This paper gives the results of experiments made by the Bethlehem Steel Company with a small experimental pump to determine the best proportions of the pump valve to be used in the Baltimore sewerage pumping engine. The experiments showed that a plain, flap valve with a seat angle of 60 deg. with the vertical plane, weighing about 0.26 lb. per sq. in. (in water) of seat opening, or 0.30 lb. in the atmosphere, measured normal to the face of the valve, will give a quiet and prompt working valve up to fully 30 r.p.m.

A discussion of the theory involved in the action of the valve, namely, that the velocity of the flow through the valve opening corresponded quite closely to the velocity due to this head or weight, held very fairly. Also that on opening, the valve lift was in perfect synchronism with the plunger velocity at any speed. In closing, the valve lagged behind for the greater part of its descent, but at the speeds given always reached its seat at the end of the stroke without shock.

THE BALTIMORE SEWERAGE PUMP VALVE

By A. F. Nagle, Bethlehem, Pa.

Member of the Society

In December 1906, the Baltimore Sewerage Commission awarded to the Bethlehem Steel Company a contract for building three sewage pumping engines of substantially the same design as the modern standard city water works pump, namely, "a self-contained, vertical, triple-expansion crank and fly wheel, three single-acting plungers, condensing engine; each pumping engine to have a capacity of 27,500,000 gal. in 24 hr., with an average total head of 72 ft. (7 ft. suction lift); and to have a capacity for 50 per cent greater speed for short periods." The only deviation from the water works design was in the pump valves. These, instead of being made of rubber discs, 3 in. or 4 in. in diameter, spring operated, were to be as follows:

2 "They shall be rectangular flap valves swinging on horizontal hinges, faced with rubber of the best quality for the purpose, and *weighted;* the valve seats shall be of hard, tough bronze, and *the planes of their faces shall be inclined.* (Italics are the author's.) The hinges of the valves shall be protected, by hooding or otherwise, from the direct flow of the sewage passing through the valves; suitable stops shall be provided to limit the opening of the valves at the extreme edges to 2½ in., and the clear waterway through either suction or discharge valve shall not be less than 145 per cent of the area of the plunger."

3 In the spring of 1907, when the drawings were well advanced, the author was appointed by the commission to be their mechanical engineer and inspector at the Bethlehem Steel Company's plant. It had been decided to make the normal speed of the engines of 5 ft. stroke and 20 r.p.m., which would be increased to 30 revolutions to comply with the contract capacity.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

This speed would necessitate a plunger 40¼ in. diameter to give the required volume of water. For the valve decks there had been tentatively planned thirty-four 4 in. x 12 in. and six 4 in. x 9 in. valves.

4 The sewage pump valve is far from being of a standard design. The commission's engineers had given the subject much study and made many inquiries before writing the paragraph quoted, and yet it is noticeable that they refused to be specific on the essential elements of the valve, namely, *its seat-angle* and *its weight*. As there would be required over 700 of these bronze valves, it would be an expensive affair to make a change in them after completion. The writer therefore suggested to the management of the Bethlehem Steel Company that they construct a small pump for the purpose of trying out one set of the proposed valves. With this they were in hearty accord, and Fig. 1 shows a sectional elevation of the pump as made. A description is hardly necessary except to say that it is 13 in. in diameter by 15 in. stroke, making from 20 to 30 r.p.m.; being the exact equivalent of 6½ in. diameter by 5 ft. stroke, the proportional

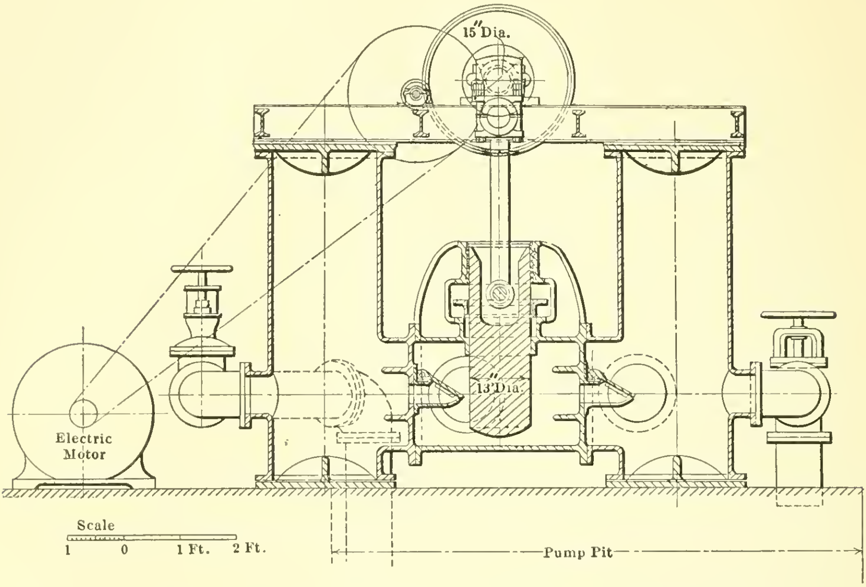


Fig. 1 Pump for studying Valves designed for Baltimore Sewage

ratio of plunger and valves. The 7 ft. suction lift was also reproduced.

5 In order to obtain definite knowledge of the action of the valve, there was attached to the arbor of its hinge a ⅝-in. shaft extending through the casing and a light stuffing box, and to this shaft were fastened suitable arms, levers and rods leading to an ordinary indicator from which the spring had been removed. In this way, a perfect trace of the lift of the valve could be obtained, examples of which accompany this paper. At the same time a pump card was taken and some of these are also reproduced.

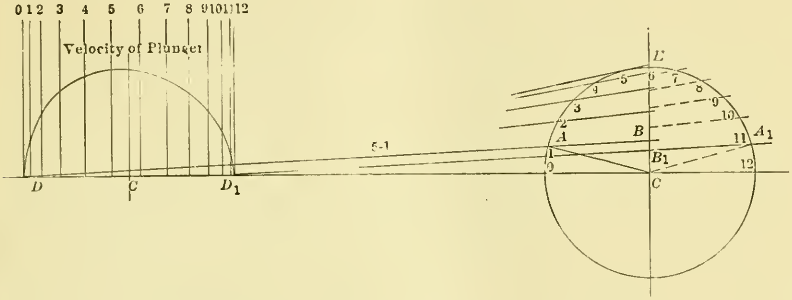


Fig. 2 Graphical Method for finding Plunger Velocity

6 The plunger velocity is dotted on the valve card, making the maximum plunger velocity agree with the highest lift of valve. The graphical method of finding this velocity is reproduced in Fig. 2, copied in substance from Rankine's Rules and Tables, page 236. The only difference in condition between this small pump and the main pump is that the relatively more rapid motion of the plunger in this case is in the lower end while in the main pump it will be at the upper end. The practical result of this arrangement is that the suction valve is favored in closing time, while in the main pump it will be the discharge valve thus favored.

7 It was decided to experiment with three different seat angles, 30, 45, and 60 deg. with the vertical plane, and weighted to different amounts. From the large number of tests made, two sets of cards for each of the three seat angles have been selected, at 20, 25, and 30 r. p. m.

8 The first test at a change of seat angle was always made with the normal or unweighted valve. Then followed one with

the suction valve loaded 5¼ lb., and the discharge valve 10.8 lb. There is no good reason, however, why the valves should not have been weighted alike.

9 Valve and pump cards are shown in Figs. 3, 4, 5, 6, 7, and 8, and data pertaining thereto are given in Table 1, Par. 23.

EXPLANATION OF TABLE 1

10 Columns 1 to 5 inclusive are self-explanatory. Columns 6 and 7, weight of valve, is the entire weight before it is put in place. Columns 8, 9, and 10, 11, are obtained by a spring scale attached to the outer edge of the flap and noting the force required to start the valve; and again, the amount at its full lift of 2½ in., the scale being held, of course, at right angles with the flap. It should be noted that owing to friction of the hinge, it usually required from one to two pounds more to move the flap upward than downward, and the amount given in the table is the mean of the two.

Full Lines represent Valve Motion - Dotted Lines represent Plunger Velocity

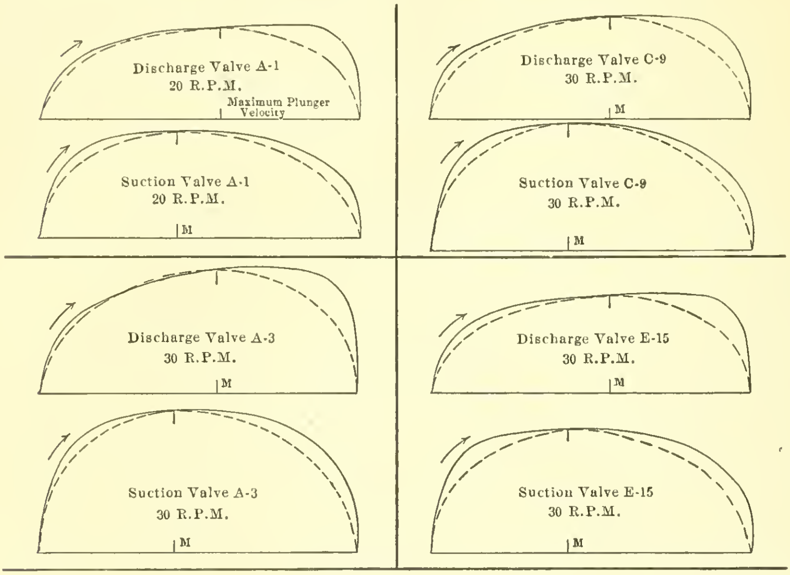


Fig. 3 Valve Cards, Seat Angle 30 Deg.

11 Columns 12-13: The actual lift of the valve is obtained as follows: Owing to the intercession of certain cranks and levers,

the pump card trace is not exactly in line with the outer edge of the flap. To correct for this the valve cards as taken must be multiplied by the following factors: 1.25 for the 30 deg. seat; 1.25 for the 45 deg. seat; and 1.54 for the 60 deg. seat.

MECHANICAL ACTION OF THE VALVES

12 Mr. A. J. Mayer, on behalf of the Bethlehem Steel Company, took all cards and made the following comments on the working of the pump valves:

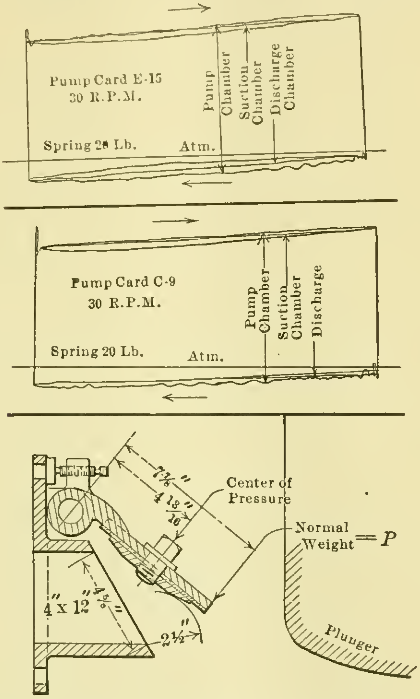


FIG. 4 PUMP CARDS, SEAT ANGLE 30 DEG.

Seat Angle, Deg.	Card No.	R.p.m.	Suction Valve	Discharge Valve
	A-1	20	Knocks slightly at closing. Very slight knock at opening.	Knocks at opening louder than suction. Very slight knock at closing.
30	A-2	25	Knocks slightly at closing. Very slight knock at opening.	Knocks at opening louder than suction. Very slight knock at closing.

Seat Angle, Deg.	Card No.	R.p.m.	Suction Valve	Discharge Valve
	A-3	30	Knocks slightly at closing. Very slight knock at opening.	Noises slightly increased at opening and closing.
30	C-7	20	Barely audible knock at closing. No noise at opening.	Pounds at opening. No noise at closing.
	C-8	25	Barely audible knock at closing. No noise at opening.	Pounds at opening. Barely audible pound at closing.
	C-9	30	Barely audible knock at closing. No noise at closing.	Noise slightly increased at opening. Very slight pound at closing.
45	21	20	No noise at opening or closing.	Knock slightly at opening and closing.
	22	25	No noise at opening or closing.	Noises slightly increased.
	23	30	No noise at opening or closing.	Knocks loud at opening and closing.
45	27	20	No noise at opening or closing.	Barely audible pound at opening. No noise at closing.
	28	25	No noise at opening or closing.	Knocks slightly at opening and closing.
	29	30	No noise at opening or closing.	Knocks slightly at opening and closing.
60	39	20	No noise at opening or closing.	Barely audible thud at opening. No noise at closing.
	40	25	No noise at opening or closing.	Audible thud at opening, and closing.
	41	30	No noise at opening or closing.	Pounds slightly at opening. Audible thud at closing.
60	45	20	No noise at opening or closing.	Barely audible thud at opening. No noise at closing.
	46	25	No noise at opening. Slight thud at closing.	Makes thud at opening and closing.
	47	30	No noise at opening. Slight thud at closing.	Makes thud at opening and closing.

Seat Angle, Deg.	VALVE DIMENSIONS	Square Inches
30	Area of Valve Seat Outside	82
	Area of Valve Seat Inside	55
	Ratio of Areas, 1 to 1.48	
	Area of Full 2½-in. Valve Lift	45
45	Area of Valve Seat Outside	101
	Area of Valve Seat Inside	72
	Ratio of Areas, 1 to 1.41	
	Area of Full 2½-in. Valve Lift	49.50
60	Area of Valve Seat Outside	107
	Area of Valve Seat Inside	80
	Ratio of Areas, 1 to 1.34	
	Area of Full 2½-in. Valve Lift	55

NOTE: It should be stated that owing to the nearness of the hub of the hinge to the body of the valve, only the front and two sides of the opening were taken as discharge area.

13 A point is now reached where there are facts to serve as a guide in the selection of a design for the valve, without any mathematical calculation whatever, the application of which is left to a later part of this paper. Referring for a moment to Mr. Mayer's remarks, attention should be called to the difficulty in distinguishing between a "thud" and a "knock" of a valve, either suction or discharge. The writer's opinion is that a "thud" always occurs at the *opening* of a valve, due to the difference between the inner and outer seat area, thus necessitating a greater pressure within to start to open the discharge valve than exists without; and the converse for the suction valve. All the pump cards show this greater or less pressure by a descent or an ascent of the indicator piston, which latter, however, falls to the normal pump pressure as soon as the valve opens. If there were air-pockets in a pump, the pressure of the air when released would send the valves violently against their stops. The soft rubber or leather facing allows this change of pressure to occur with nothing worse than a "thud."

14 In addition to the valve action, there is at this time the reversal of all stresses in the moving parts and journals, thus tending to increase the "thud" of the pump.

15 In contrast with a "*thud*" at *opening* is the "*knock*" at *closing*. This knock is due to the violent seating of the valve by the reversed action of the plunger, if, instead of coming gently

to its seat by its inherent weight as the plunger reaches the end of its stroke, it is not then seated.

SELECTION OF A DESIGN

16 A cursory examination of the valve traces or cards will show:

First, that *at opening*, every valve trace followed closely the theoretical or plunger-velocity line, except those of 30-deg. seat angle. In this case the valve acted like a swinging pendulum, in the impetus it received in opening, notwithstanding its natural increased weight, due to change of angle, carrying it beyond its proper position. In studying these cards, one must not lose sight of the different plunger velocities at the two ends of the stroke.

Full Lines represent Valve Motion-Dotted Lines represent Plunger Velocity

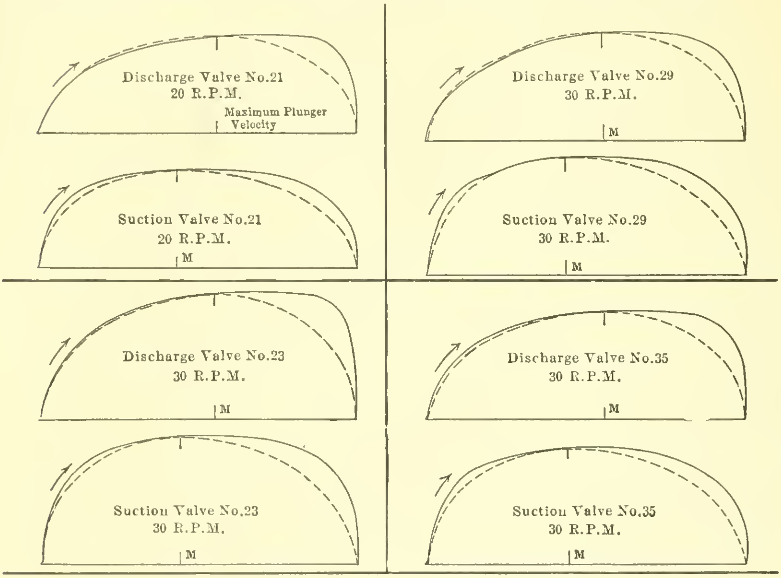


Fig. 5 Valve Cards, Seat Angle 45 Deg.

Second, in closing there was a constant improvement from the 30 deg. to 60 deg. seat angle; and, of course, with its increased weight. The heavier the valve the better it worked, mechanically, but the lift was reduced—

not desirable in a sewage valve—and the velocity through the valve increased.

17 In opening, every pump valve, either suction or discharge, if of constant or nearly constant weight, such as all gravity acting valves are, is *forced* by the plunger to follow its own varying velocities; but in *closing* there is no force or pressure to do this.

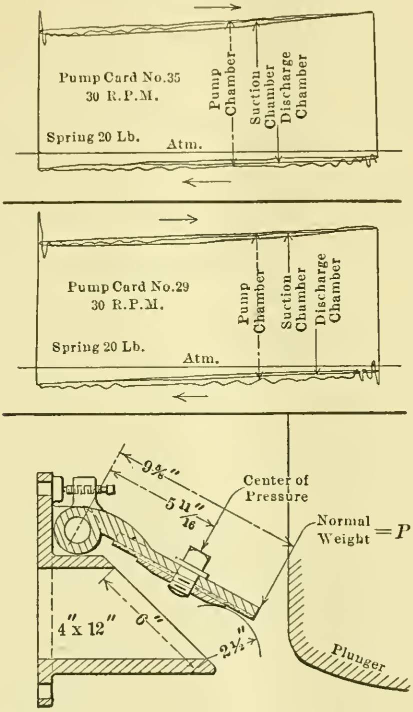


Fig. 6 Pump Cards, Seat Angle 45 Deg.

It has only its own inherent weight to move it, and its effective weight is diminished by several causes. First, there is the resistance of its hinge, probably very small, since the fit is very free; second, the water cannot fill in the back of a flat valve any more than it can flow into the space left by the square stern of a boat; hence the partial vacuum formed there increases the lifting force of the stream beneath it; and third, there is the inertia of the mass of the valve as well as of the water which has to change its position.

18 Perhaps the experience all have had in rowing a boat up or down a rapidly moving stream will give a good impression of the difference between closing and opening a pump valve.

19 At 30 revolutions the 60 deg. seat valve (unweighted) reached only 2 in. of its 2½ in. possible lift. To have made it light enough to go to its 2½-in. stop at this speed would have necessitated reducing its weight about one-third (see last line in Table 5) and with every indication that it would not reach its seat in closing in a way to prevent slamming. Valve cards 39-41 furnished the best proportions for a large lift and lowest velocity through the valve. Its weight of seat area per square inch was 0.221 lb. (See Table 3.) The extraordinary lift of 2½ in. of the suction valves 41 and 47 can be attributed to the proximity of the plunger at the lower end of its stroke, thus causing a back pressure at the valve outlet.

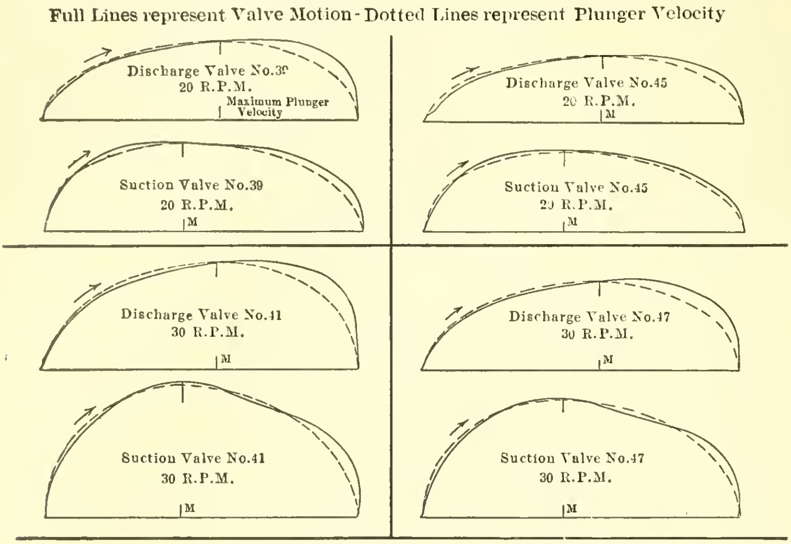


Fig. 7 Valve Cards, Seat Angle 60 Deg.

20 Fig. 9 is a reduced scale drawing of the valve as actually constructed. It weighs about 0.30 lb. per sq. in. of its seat area. It is of bronze throughout and has a renewable leather face. Its stop is cast on the body of the valve above it. There are 44 valves exactly alike, grouped into a 7-ft. steel valve deck. Figs. 11 and

12 are reduced photographs of these decks. Fig. 11 shows the valves closed and Fig. 12 the valves blocked open. Fig. 13 shows the pump end of the engine as it was erected in shop No. 2 of the Bethlehem Steel Company. The writer standing at the base, furnishes a scale to the pumps.

21 These pumps have now been in operation for several months and from the first turnover have worked so quietly one would hardly know there were any valves in them.

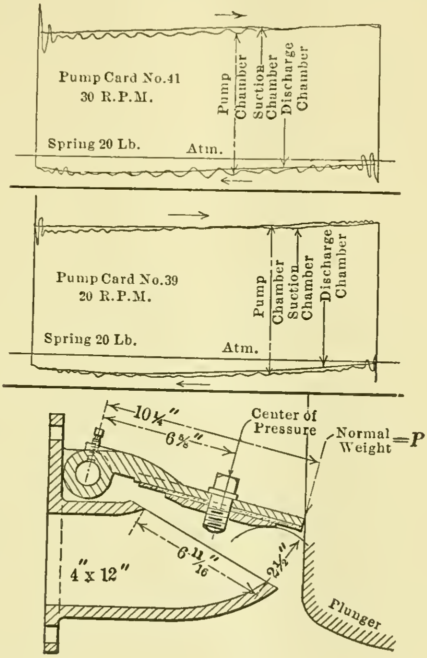


FIG. 8 PUMP CARDS, SEAT ANGLE 60 DEG.

22 Aside from the design of the valve itself, great care was taken that all valves should be of like weight. When a group of valves are thus tied together, each is held in a very sensitive balanced position and any difference in weight is reflected in a different lift; hence a single light valve may be held back in closing and then slam to its seat. This was actually demonstrated by the action of the air-relief valve on top of the pump, which is a common swivel, swing check valve, without stop or definite weight.

23 Having described the valve as constructed, and conscious of its excellencies, shortcomings, cost of production and possible cost of renewals, the writer ventures to suggest that a cast-steel flap on a rounded seat would wear longer, cost much less to produce or repair and work fully as well, without a "thud," if the seat were of bronze or possibly of cast iron. Half a dozen of these valves, as shown in Fig. 10, have been made for future trial.

TABLE 1 PUMP VALVE EXPERIMENTS. OBSERVED DATA

Seat Angle, Deg.	Card No.	R.P.M.	Pump Head, Ft.			Weight of Valve, Lb.		Normal Weight of Valve				Lift of Valve	
								Suction		Discharge			
			Suction	Discharge	Total	Suction	Discharge	Start	Top	Start	Top	Suction	Discharge
	1	2	3	4	5	6	7	8	9	10	11	12	13
30	*A*-1	20	7.59	67.1	74.7	27	28	8	11	8	10.5	1.60	1.38
30	*A*-2	25	7.71	65.8	73.5	27	28	8	11	8	10.5	1.85	1.63
30	*A*-3	30	7.93	66.9	74.8	27	28	8	11	8	10.5	2.15	1.90
30	*C*-7	20	7.25	66.9	74.2	32.2	38.8	11.5	14.5	14.0	17.5	1.46	1.14
30	*C*-8	25	7.71	65.6	73.3	32.2	38.8	11.5	14.5	14.0	17.5	1.65	1.29
30	*C*-9	30	7.93	65.0	72.9	32.2	38.8	11.5	14.5	14.0	17.5	1.92	1.53
45	21	20	6.35	65.9	72.2	30	31	10.1	11.1	10	11	1.50	1.50
45	22	25	6.35	66.7	73.0	30	31	10.1	11.1	10	11	1.70	1.69
45	23	30	6.35	64.6	70.9	30	31	10.1	11.1	10	11	1.96	1.94
45	27	20	7.03	67.1	74.2	35.2	41.8	13.7	15.2	16.7	19.5	1.36	1.24
45	28	25	7.03	67.9	74.9	35.2	41.8	13.7	15.2	16.7	19.5	1.50	1.38
45	29	30	7.03	66.8	73.8	35.2	41.8	13.7	15.2	16.7	19.5	1.80	1.64
60	39	20	6.80	67.3	74.1	34	32	13.2	14.2	12.7	13.6	1.69	1.48
60	40	25	6.80	66.1	72.9	34	32	13.2	14.2	12.7	13.6	2.20	1.81
60	41	30	6.80	66.1	72.9	34	32	13.2	14.2	12.7	13.6	2.50	2.00
60	45	20	6.80	64.5	71.3	39.2	42.8	17.6	19.1	21.1	22.4	1.54	1.25
60	46	25	6.80	65.6	72.4	39.2	42.8	.....	.....	.....	.....	1.92	1.51
60	47	30	7.36	66.1	73.4	39.2	42.8	.....	....	.....	.....	2.26	1.68
			Aver.......		73.3								

AGREEMENTS

24 Having described the essential facts leading to the selection of the design of this sewage pump valve, the writer does not like to leave the subject without an effort to present a theory as a guide to the construction of similar valves. Aside from the evidence given by the valve cards of the better working of the 60-deg. seat valve than of the valves with 30 deg. or 45 deg. angles,

there is the important result of a much larger outlet area with the same lift of valve, and consequent lower velocity through the valve. This is shown in the following tabulation of results with the unloaded valves at 30 revolutions:

30 deg. seats 34.2 sq. in. outlet, 1.90 lift, 7.72 ft. velocity
45 deg. seats 38.6 sq. in. outlet, 1.95 lift, 6.48 ft. velocity
60 deg. seats 44.4 sq. in. outlet, 2.00 lift, 5.94 ft. velocity

TABLE 2 PUMP VALVE EXPERIMENTS. CORRECTIONS OF ORIGINAL DATA APPLIED TO DISCHARGE VALVE. SEE FIGS. 14 AND 15

Seat Angle, Deg.	Card No.	R.P.M.	Lift of Valve		Normal Weight of Valve			
			Log, In.	Corrected In.	Log		Corrected	
					Start	Top	Start	Top
	1	2	3	4	5	6	7	8
30	*A*-1	20	1.38	1.43	8	10.5	8	10.5
30	*A*-2	25	1.63	1.61	8	10.5	8	10.5
30	*A*-3	30	1.90	1.90	8	10.5	8	10.5
30	*C*-7	20	1.14	1.14	14	17.5	14	17.5
30	*C*-8	25	1.29	1.29	14	17.5	14	17.5
30	*C*-9	30	1.53	1.54	14	17.5	14	17.5
45	21	20	1.50	1.46	10	11	10	11.7
45	22	25	1.69	1.68	10	11	10	11.7
45	23	30	1.94	1.95	10	11	10	11.7
45	27	20	1.24	1.20	16.7	19.5	16.7	19.5
45	28	25	1.38	1.38	16.7	19.5	16.7	19.5
45	29	30	1.64	1.61	16.7	19.5	16.7	19.5
60	39	20	1.48	1.51	12.7	13.6	12.7	13.6
60	40	25	1.81	1.75	12.7	13.6	12.7	13.6
60	41	30	2.00	2.00	12.7	13.6	12.7	13.6
60	45	20	1.25	1.26	21.1	22.4	21.1	22.4
60	46	25	1.51	1.47	21.1	22.4	21.1	22.4
60	47	30	1.68	1.68	21.1	22.4	21.1	22.4

In the latter case, the throat area of 48 sq. in. is very closely approached. (See Fig. 17.)

25 The basis of all hydraulic calculations is the feet head, or pounds pressure, per square inch of area, and consequent velocity of flow in feet per second. This relation is fixed absolutely by natural law and if theoretical results are not obtained, one must look for some diverting cause, such as contraction, change of direction, or friction. These last effects must be ascertained by experiment. For simple forms such as pipes, the values are

well established; but for irregular shapes such as these valve openings present, no data exist, and from the limited amount of information gathered during the experiments, the writer is unable to make a complete analysis of the valves. They do give *something*, however, as a guide in future construction, which the writer will endeavor to present.

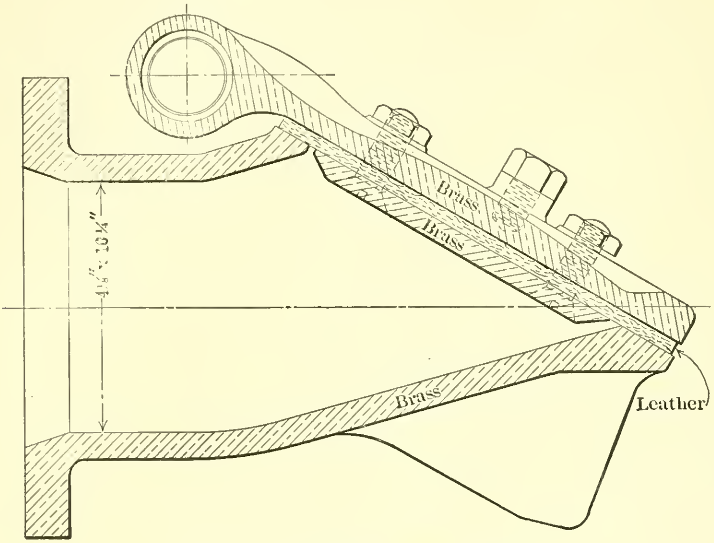


Fig. 9 Valve as constructed

26 The problems which it was attempted to solve are these:

First, knowing the weight—the normal weight—of a valve per square inch of seat area, how nearly will that weight, or pressure, give a velocity through the valve in agreement with the actual velocity found by experiment?

Second, knowing the lift of a valve at a given number of revolutions, how much will this lift be increased by an increase of revolutions?

Third, knowing the lift and weight of a valve at a given speed, what will be the lift if the weight be increased but the speed remain the same?

27 Before proceeding with the analysis it should be explained that it will be confined to the discharge valve only, and for these

reasons: In the first place, the writer has no theory of valve action that would distinguish between the two; second, certain discrepancies in the data, especially relating to the suction valve, led him to doubt their accuracy; and, third, there is no use in burdening the paper with more figures than are absolutely necessary.

28 As already explained, the original or log figures are given in Table 1. For the purpose of the calculations it has seemed necessary to make certain corrections upon these, as shown in Table 2 and in diagrams Figs. 14 and 15. Numerous small changes were made in the lift of the valve, but only one in the normal weight of the valve. These corrected values were used in all subsequent calculations, as exhibited in Table 3.

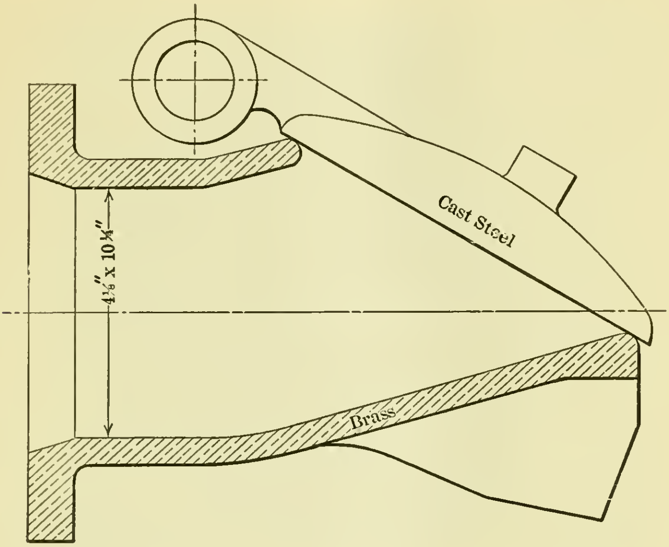


Fig. 10 Design of Valve proposed by the Author

EXPLANATION OF TABLE 3

29 Column 14: The maximum plunger velocity is really slightly more than the crank velocity—for a 5 to 1 connecting rod it amounts to nearly 2 per cent, or 1.60 times the plunger travel, and this is given in this column. The main engine dimensions have been preserved rather than those of the experi-

mental pump, no change in calculated result being affected thereby.

30 Column 15: To facilitate subsequent calculations, there is given the plunger area corresponding to each valve (which is 33.00 sq. in.) multiplied by its velocity, giving the discharge per second in inch-feet.

31 Column 16: The area of valve opening is found by taking the full area opening given under "Valve Dimensions" in Par. 12, for $2\frac{1}{2}$-in. lift, and reducing it to its proportionate amount for the actual lift given in Table 2, column 4.

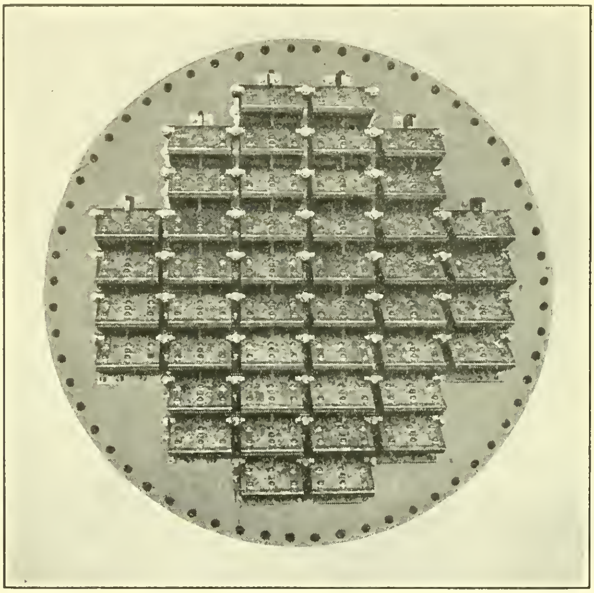


Fig. 11 View of Valves, Closed

32 Column 17: The apparent velocity through the valve is an important result. It is found by dividing the rate of discharge of plunger given in column 15, by the area of valve opening given in column 16. Therefore, if any two adjoining values in columns 16 and 17 be multiplied the corresponding result given in column 15 will be obtained. This is called the *apparent velocity*, although commonly called the *actual velocity*, for it is what the velocity would be if the area given were wholly filled with the issuing stream; but there must have been some con-

traction, and hence a correspondingly higher velocity. The writer is unable to divide this apparent velocity into the various measured causes that produce it, and with this explanation rests this part of the discussion because he is unable to carry the analysis any further with the data at hand.

33 Column 18: Hydraulic engineers always desire to express the relation of *head* to *velocity*. Hence, if in column 17 the velocity through the valve is given, what is its corresponding head? Reducing the standard formula for feet head to pounds pressure, he has used the formula

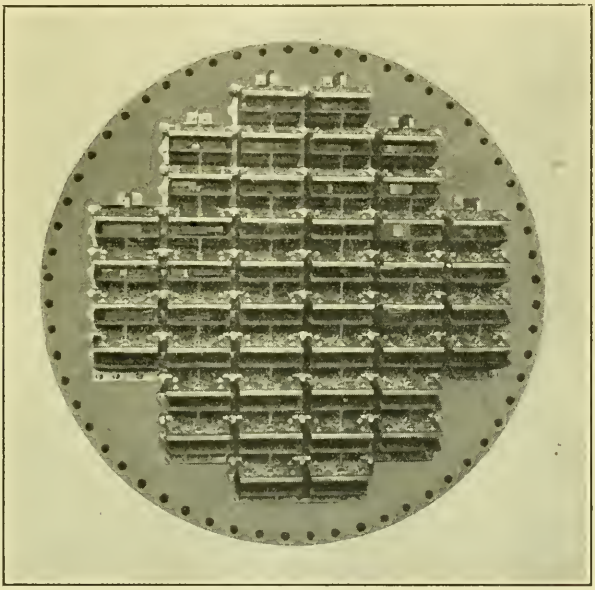


FIG. 12 VIEW OF VALVES, OPEN

$$\text{Vel.} = 12.20 \sqrt{P} \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

P being the pressure in pounds per square inch.

34 Column 19: The normal weight of valve is given in Table 2, only at its two extreme positions. It is necessary to know what it is at the position of its lift. If, for example, it takes 8 lb. to lift valve A-1 at the start, and 10.5 lb. at its 2½-in. lift, the weight increases just one pound for each inch of lift. Hence, if the lift is 1.43 in. the weight at 1.43 in. lift will be 9.43 lb. These values are found in this column.

35 Column 20: The normal weight of valve per square inch of seat area is found as follows: In the previous column (19) is given the total normal weight at *the edge of the valve.* Taking the center of pressure of the issuing stream at the center of the seat opening (which is probably not exactly correct, but which is the best that can now be affirmed), it is needed but to reduce the extreme total weight to this center, divide by the seat area in square inches and correct for loss of weight in water.

TABLE 3 PUMP VALVE EXPERIMENTS. CALCULATIONS BASED UPON OBSERVED DATA APPLIED TO DISCHARGE VALVE ONLY

Seat Angle, Deg.	Card No.	R.P.M.	Max. Plunger Velocity, Ft. per Sec.	Rate of Max. Pump Discharge, In-Ft. per Sec.	Area of Valve Opening, Sq. In.	Apparent Velocity through Valve, Ft. per Sec.	Velocity Head due to Velocity through Valve, Lb.	Normal Weight of Valve at Lift, Lb.	Normal Weight of Valve per Sq. In. of Seat Area, Lb.	Velocity due to Normal Weight of Valve, per Sq. In. of Seat Area, Lb.	Ratio of Velocities of Col. 21 to Col. 17
	1	2	14	15	16	17	18	19	20	21	22
30	*A*-1	20	5.33	176	25.7	6.85	0.315	9.43	0.247	6.08	1.13
30	*A*-2	25	6.66	220	29.0	7.59	0.387	9.61	0.252	6.12	1.24
30	*A*-3	30	8.00	264	34.2	7.72	0.400	9.90	0.260	6.22	1.24
30	*C*-7	20	5.33	176	20.5	8.58	0.494	15.6	0.409	7.81	1.11
30	*C*-8	25	6.66	220	23.2	9.48	0.603	15.8	0.415	7.87	1.21
30	*C*-9	30	8.00	264	27.7	9.53	0.610	16.2	0.425	7.95	1.20
45	21	20	5.33	176	28.9	6.09	0.249	11.0	0.222	5.75	1.06
45	22	25	6.66	220	33.7	6.53	0.286	11.2	0.224	5.77	1.13
45	23	30	8.00	264	38.6	6.48	0.282	11.3	0.229	5.84	1.11
45	27	20	5 33	176	23.8	7.39	0.366	18.0	0.363	7.35	1.01
45	28	25	6.66	220	27.0	8.15	0.436	18.2	0.367	7.39	1.10
45	29	30	8.00	264	31.9	8.28	0.460	18.5	0.373	7.45	1.11
60	39	20	5.33	176	33.2	5.30	0.189	13.2	0.219	5 71	0.93
60	40	25	6.66	220	38.5	5.71	0.219	13.3	0.221	5.74	1.00
60	41	30	8.00	264	44.4	5.94	0.237	13.4	0.223	5.76	1.03
60	45	20	5.33	176	27.7	6.35	0.270	21.8	0.363	7.35	0.86
60	46	25	6.66	220	32.3	6.81	0.311	21.9	0.364	7.36	0.90
60	47	30	8.00	264	37.0	7.14	0.342	22.0	0.366	7.38	0.94

36 For the 30 deg. seat are these values, *P* being the total normal weight at lift, found in column 19:

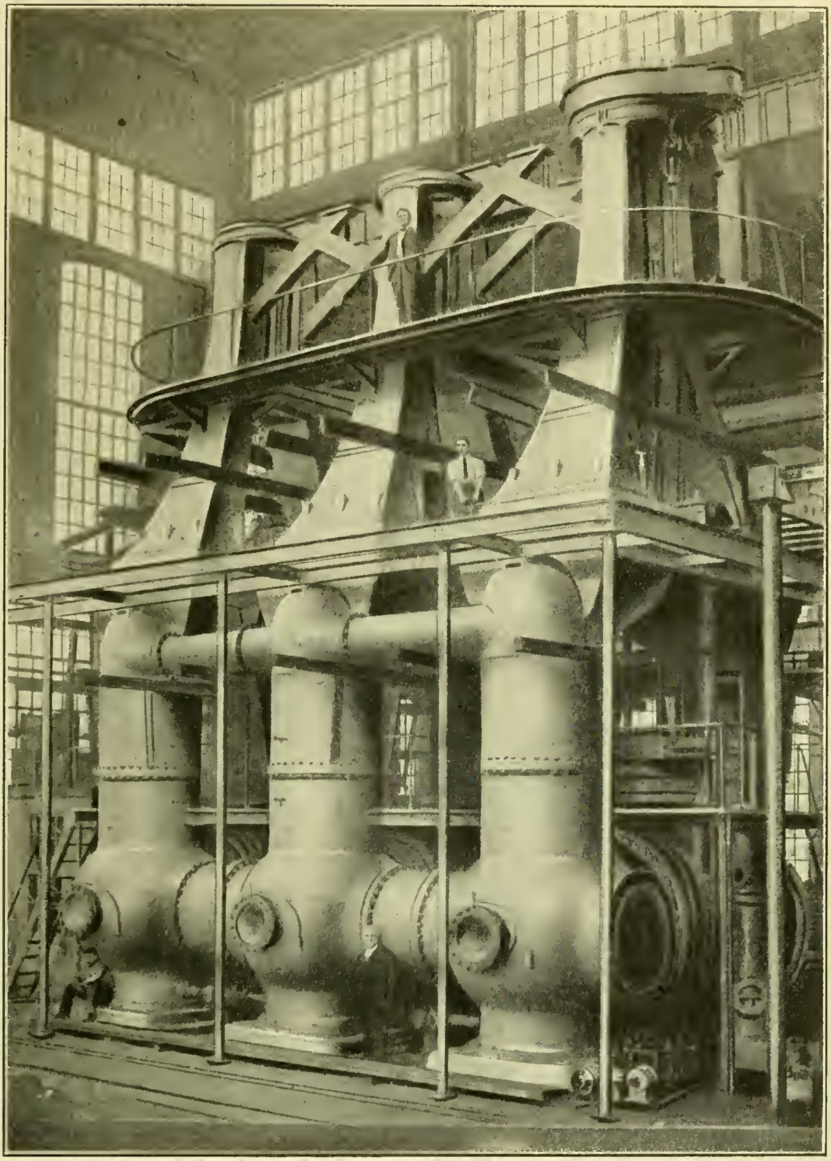


Fig. 13 Pumps of the Baltimore Sewage Pumping Engine as erected in the Shop

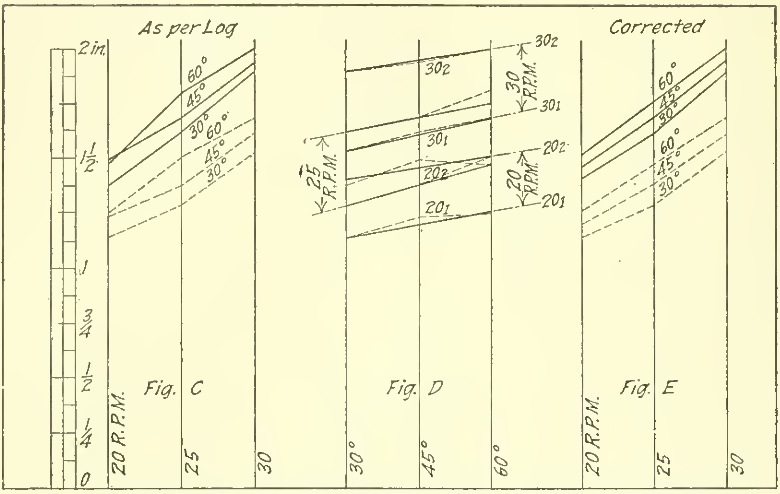
$$W = \frac{7.87 \times 0.86 \times P}{4.81 \times 0.55} = 0.0256\ P \ldots\ldots\ldots\ldots [2]$$

For the 45 deg. seat these values become

$$W = \frac{9.62 \times 0.86 \times P}{5.69 \times 0.72} = 0.0202\ P \ldots\ldots\ldots\ldots [3]$$

For the 60 deg. seat

$$W = \frac{10.2 \times 0.86 \times P}{6.62 \times 0.80} = 0.0167\ P \ldots\ldots\ldots\ldots [4]$$



Note: The Full or Upper Group of Lines is for the Unweighted Valve
" Dotted " Lower " " " " " " Weighted "

Fig. 14 Lift of Discharge Valve, corrected. See Table 2

37 Column 21: *Velocity* due to weight of valve in water per square inch of seat area, as given in column 20, is found by formula [1].

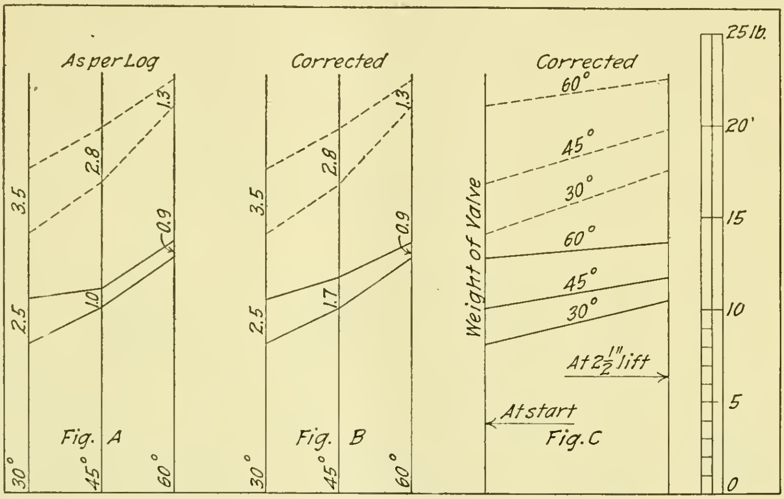
38 Column 22: It now remains to see how closely these calculated velocities agree with the apparent velocities as given in column 17. These ratios are found in this column, and for better study they are plotted in Fig. 16.

39 A check on the ratios of velocities in column 22 can be had by finding a similar ratio of the weight of the valve and weight or " head " due to the velocity through the valve, as given in columns 18 and 20, bearing in mind, however, that the velocities vary with the square root of the pressure or " head." For

example—taking the first line in Table 3, columns 18-20, the weights are respectively as 0.315 to 0.247, and their square roots as 0.561 to 0.500, or 1.12 to 1, practically the same as found in column 22; these slight divisions being due to the number of decimals carried in the calculations.

COMPARISONS

40 The point has now been reached where the results of these experiments can be considered with some degree of intelligence.



Note: Full Lines Represent Unweighted Valve
Dotted " " Weighted "

FIG. 15 TOTAL NORMAL WEIGHT OF VALVE AT START AND AT 2½ IN. LIFT. SEE TABLE 2

Turning to the first proposition propounded, How nearly does the calculated velocity, based upon the method here described, agree with the actual velocity through the valve as found by these experiments? On the whole, it seems to do so very closely; indeed, in the case of the 60 deg. seat and light valve, almost exactly. Rather than attempt to give mathematical expression to these deviations and apparent discrepancies, the designing engineer will use his own judgment as to the proper allowance to be made for them. Sufficiently accurate information has been obtained to prevent future mistakes in the construction of this type of valve as applied to the described conditions.

41 Second. In what proportion does the lift of the valve change with the speed? In Table 4 these results are tabulated and reduced to percentages, where they can be studied. The comparatively rough and uneven covering plate, holding the leather in place, is largely responsible for these erratic deviations. With a smooth valve face, as shown in Fig. 10, the deviations would probably have been gradual and proportionate to the change of angle by the lift, and a value might be given to them for different seat angles, but that will not be attempted here.

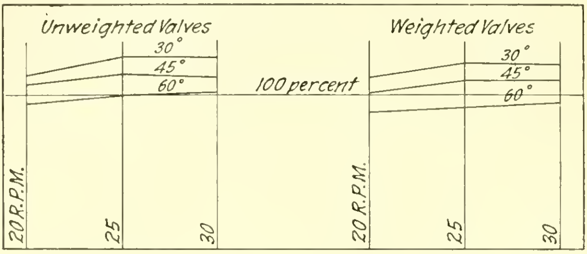


Fig. 16 Ratio of Agreement between Calculated and Actual Velocities through the Valve. See Col. 22, Table 3

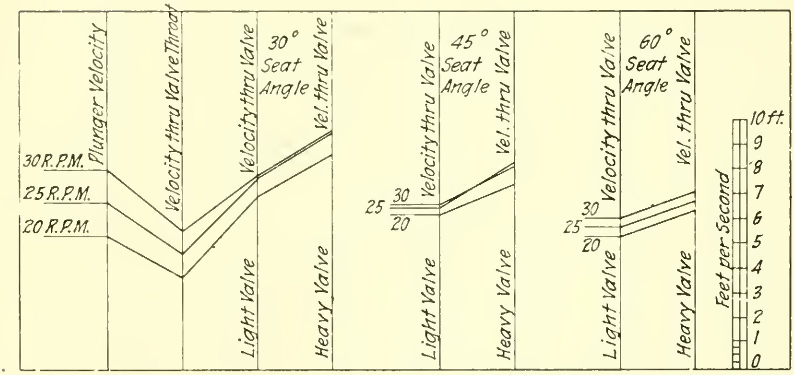


Fig. 17 Maximum Velocity, Feet per Second, of Plunger and of Flow through Throat of Valve and through Discharge Valve at 20, 25, and 30 R.P.M. and with 30, 45 and 60 Deg. Angle of Valve Seat

42 Third. The speed remaining the same, how does a change of weight affect the lift of the valve? Table 5 in its first six columns groups together previously given data, and in column 7 is applied the theoretical law that the lift should be inversely pro-

portional to the square root of the weight of the valve. That is, taking the lighter valve and its lift, what should be the reduced lift by loading it? In every case, it will be observed, the valves lifted *higher* than they should, according to this rule. On the 30 deg. seat, this variation was only about 3 per cent; on the 45 deg. seat, about 5 per cent; and on the 60 deg. seat, about 7.6 per cent. Why should it have lifted higher than calculated? The only explanation seems to be that the increased weight reduced the angle of impingement of the issuing stream against the face of the valve, and that this change of angle was proportionately more on the 60 deg. seat than on the others.

TABLE 4 COMPARISON OF LIFT OF VALVE WITH INCREASED SPEED

THEORETICALLY, THE LIFT SHOULD BE DIRECTLY PROPORTIONAL TO THE SPEED

Seat Angle, Deg.	R.P.M.	Light or Unweighted Valve				Heavy or Weighted Valve			
		Actual, In.	Theoretical, In.	Deviation		Actual, In.	Theoretical, In.	Deviation	
				Per Cent	Per Cent			Per Cent	Per Cent
1	2	3	4	5	6	7	8	9	10
30	20	1.43	1.43	100	...	1.14	1.14	100	...
30	25	1.61	1.79	90	100	1.29	1.42	91	100
30	30	1.90	2.14	88	98	1.54	1.71	90	100
45	20	1.46	1.46	100	...	1.20	1.20	100	...
45	25	1.68	1.82	92	100	1.38	1.50	92	100
45	30	1.95	2.19	89	96	1.61	1.80	89	97
60	20	1.51	1.51	100	...	1.26	1.26	100	...
60	25	1.75	1.89	92	100	1.47	1.57	94	100
60	30	2.00	2.26	88	95	1.68	1.89	88	95
	40	2.50				2.50	47 revolutions		

NOTE.—These last valves would have gone to their stops at 40 and 47 r.p.m. respectively.

NOTES

43 In Fig. 18 are shown the areas of the plunger displacement and the valve areas. At once one may ask why these enlarged valve areas are greater than that of the valve opening, or outlet; or perhaps more pertinently, Why does not the engine

builder open the valve so as to give this required area when specifications call for, say, valve area of 145 per cent of plunger area? Simply because he cannot do it. If the valve were made light enough to open to this height, it would open, but it would close with a slam that would sound like a 10-in. gun explosion.

44 In Fig. 17 are plotted the plunger velocities and velocities through throat and outlet of valve, and it will be observed that

TABLE 5 COMPARISON OF WEIGHT AND LIFT OF VALVE

The Lift should be Inversely Proportional to the Square Root of its Weight. Discharge Valve only

Seat Angle, Deg.	R.P.M.	Weight per Sq. In.		Lift of Valve			
		Unweighted, Lb.	Weighted, Lb.	Unweighted, In.	Weighted, In.	According to Rule	Deviation, Per Cent
1	2	3	4	5	6	7	8
30	20	0.247	0.409	1.43	1.14	1.11	1.026
30	25	0.252	0.415	1.61	1.29	1.25	1.028
30	30	0.260	0.425	1.90	1.54	1.49	1.036
45	20	0.222	0.363	1.46	1.20	1.14	1.050
45	25	0.224	0.367	1.68	1.38	1.31	1.051
45	30	0.229	0.373	1.95	1.61	1.52	1.054
60	20	0.219	0.363	1.51	1.26	1.17	1.074
60	25	0.221	0.364	1.75	1.47	1.36	1.078
60	30	0.223	0.366	2.00	1.68	1.56	1.076
60	30	0.165	.....	2.50	....	....	.....

Note.—This last valve, weighing 0.165 lb. per sq. in. of seat area, making 30 r.p.m. would have gone to its 2½ in. stop.

with the lightest valve, on a 60-deg. seat, the velocity through the valve reached 5.30 ft. a second, while the plunger made 5.33 ft. a second. And yet, late water works specifications contain the clause that " the velocity through the valve shall not exceed 3 ft. a second! " Such a low velocity was never obtained through a pump valve. It would call for a valve weighing only 0.06 lb. per sq. in. of seat area—1/10 of the actual pressure or weights of valves, in practice. The writer has weighed numerous valve springs on modern city pumping engines and found them to be about 0.60 lb. (more and less) per sq. in. of net seat area at the *start*, and tightening up to as much as 1.50 lb. when lifted to a stop only 9/16 in. high, entailing a velocity of at least 9 ft. to 15 ft. a second.

45 Turning to valve card No. 41, it will be seen that it closes promptly even at 30 r. p. m. with a 2 in. lift, and its weight at the start was only 0.212 lb. per sq. in. and 0.227 lb. at its full 2½ in. lift, an increase of weight of only 7.1 per cent. This is an initial weight of only about one-third and a final weight of only about one-sixth as much as is put into spring-operated pump valves.

TABLE 6 VALVE RESISTANCE BY INDICATOR CARDS AND BY CALCULATION. SEE FIG. 19

Seat Angle, Deg.	R.P.M.	Unweighted		Weighted	
		By Indicator Per Cent	By Weight of Valve Per Cent	By Indicator Per Cent	By Weight of Valve Per Cent
1	2	3	4	5	6
30	20	2.75	2.31	4.10	3.83
30	25	3.09	2.37	3.76	3.89
30	30	4.29	2.38	2.91	3.98
45	20	2.85	2.08	3.79	3.40
45	25	3.46	2.10	4.12	3.44
45	30	4.17	2.15	4.49	3.50
60	20	2.95	2.05	3.45	3.40
60	25	3.27	2.07	2.95	3.41
60	30	2.63	2.09	3.44	3.43
Averages................		3.27	2.18	3.67	3.59

VALVE RESISTANCE

46 It will be noted that the pressure line in both suction and discharge chambers was traced on the pump indicator cards. From these, the valve resistance commonly called friction could have been obtained if an average of many cards could have been had; but the few taken for each experiment were too few to give rational results. In Table 6, columns 3 and 5 are copied from Mr. Mayer's records and plotted in Fig. 19. The fluctuations are too great for any reasonable deductions.

47 The writer has therefore attempted to make a calculation of this resistance based upon the weight of the valve which, it seems to him, is the source of this loss. Practically, all other resistances within a pump are of relatively small amount, and hence the weight of a valve furnishes a reasonable criterion of pump losses.

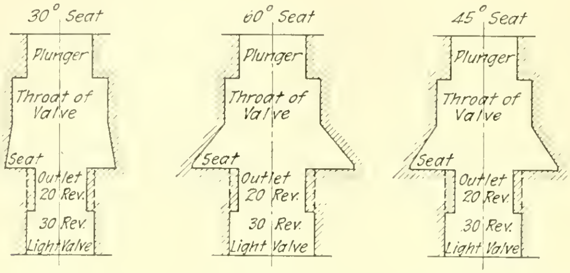


Fig. 18 Proportional Areas at Maximum Velocities

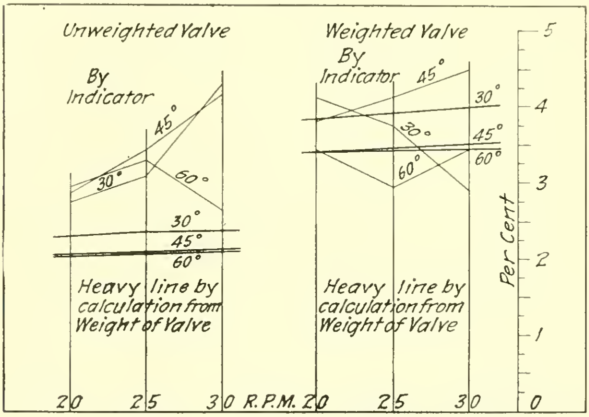


Fig. 19 Percentage of Valve Friction to 72 Ft. Head

48 It will be recalled that the apparent velocity through the valves is given in column 17, Table 3, and that no further allusion was made to the effect of contraction through the valve. If a calculation for loss of power is to be made, its amount will depend upon the amount of contraction, for the velocity must then be increased in an inverse ratio to the contraction. For the purpose of illustrating the method followed, a contraction of 81

per cent has been assumed. That would increase the velocity 23 per cent and the velocity head 50 per cent. Taking the weight of the valve as given in column 20 and increasing it by 50 per cent, gives a fair basis for calculating the loss of power within the pump. To bring this to a percentage basis it must be multiplied by two to apply it to both strokes and divided by 32 lb. for short (31.74 lb.) being the net total head in column 5, Table 1, and these values are given in columns 4 and 6, Table 6, and plotted in Fig. 19.

49 It must be left to the reader to decide whether the writer's assumptions and deductions are rational and justifiable.

50 The writer wishes to express his obligations to Mr. E. G. Grace, General Manager, Bethlehem Steel Company, for permission to make public the technical information obtained by these experiments.

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THE VAUCLAIN DRILL

By A. C. Vauclain
and
H. V. Wille

ABSTRACT OF PAPER

In the usual type of drill there are four distinct cutting edges. The actual cutting, however, is done mainly by the two edges extending from the outer circumference to points near the center. At the central portion of the drill are the two shorter cutting edges constituting the "chisel point" which do not cut freely and introduce stresses tending to split the drill.

The paper describes a new type of twist drill which has two radial cutting edges and in which the usual chisel point of such drills is eliminated.

In this drill the stresses due to the chisel point are absent and the arrangement of the cutting edges is such that a greater thickness of metal is secured at the center of the drill, adapting it to heavy feeds by which means the most economical results are secured in drilling.

Tables are given showing the results of comparative tests with the Vauclain and regular types of drills, indicating a saving with the former of over half the horsepower expended with the latter type.

THE VAUCLAIN DRILL

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Non-Member[1]

and

Henry V. Wille, Philadelphia, Pa.
Member of the Society

Speaking generally, there can be no better definition of economical drilling than "rapid drilling"—the saving of time. The fact that a drill will cut at some phenomenal speed or will consume such and such an amount of power means nothing so far as productive capacity is concerned. The object in view is the removal of chips.

2 The productive possibilities of any metal-cutting tool are limited by its stress and heat resisting capacities. The temperature at which it will continue to operate successfully depends upon the excellence of material and manufacture, but the rapidity of heat generation and the stresses set up in the tool depend upon the design of the tool and the selection of feeds and speeds.

3 With a given excellence of material and manufacture, the strength of the tool must depend upon the extent and disposition of the section. The size of the drill is necessarily limited by the size of the hole to be drilled. The section of the drill has a lesser area than that of the hole, since space must be provided for the discharge of chips from the hole. The design of the drill should therefore be that which will give it the maximum of strength and strength conservation.

4 Cutting stress is practically independent of the cutting speed and with a given feed is proportional to the lip angle of

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the cutting edge. The cutting stress does not increase as rapidly as the feed.

5 The rapidity with which cutting heat is generated depends upon the cutting speed, the depth of feed and the lip angle of the cutting edge.

6 Since both the stress and heat are influenced by the keenness of the cutting-edge, it is desirable that the lip angle be as small as possible. But it must be blunt enough to carry off the heat and to support the chip pressure, which falls more or less back of the actual cutting edge, according to the depth of feed. Since the chip is torn, not cut from the work, rupture between work and chip precedes the actual cutting edge. The heavier the feed, the farther back from the cutting edge will its pressure fall upon the tool.

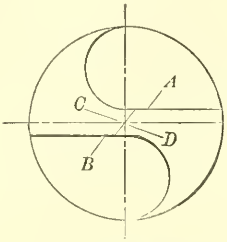

Fig. 1 Common Type of Twist Drill

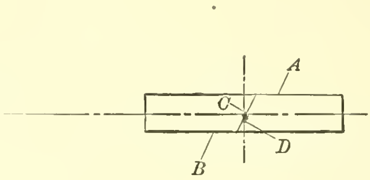

Fig. 2 Common Type of Flat Drill

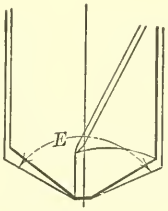

Fig. 3 Showing Included Angle of Cutting Edges of 118 Deg.

7 Under ideal conditions, the torsional capacity of the drill should be its limit of cutting strength. This does not obtain in the present commonly used types of drill, and they break down very considerably below their torsional strength. Why this is so will be explained later in detail.

8 The feed remaining constant, the horsepower consumption

will be proportional to the speed. This is true both of the power consumed by the machine and that consumed in cutting. The speed remaining constant, the power consumed in cutting does not increase as rapidly as the feed and the power consumed by the machine remains constant for all feeds.

9 From this, it will be seen that the most economical method of chip production is by giving preference to the feeds, rather than the speeds. Power, time and drills will be saved thereby.

COMMONLY USED TYPES OF DRILLS

10 Figs. 1 and 2 illustrate the section scheme of drills now commonly used. While there are many modifications of these, the figures suffice to illustrate their common characteristic, which is that the cutting edges *A* and *B* pass to one side of the axis of motion of the drill instead of through the axis. In this respect there is no difference between Figs. 1 and 2.

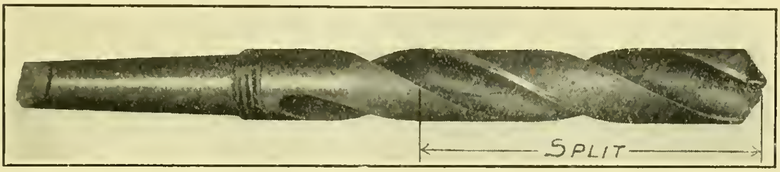

Fig. 4 Ordinary Type of Drill showing how Drill Splits Centrally in Heavy Usage

11 It will be seen that in this scheme the drill has four distinct edges, *A*, *B*, *C* and *D*, and that the usual name given to it of "two-lip" drill is not correctly applied.

12 Referring to Fig. 3, it is customary for the included angle *E* to be of 118 deg. and the cutting edges *C* and *D*, Figs. 1 and 2, therefore have an unfavorable lip angle. These edges constitute what is commonly called the chisel point and their cutting resistance is very great. The cutting edges *A* and *B* cut more freely than the cutting edges *C* and *D* and a tendency to longitudinal fracture of the drill is set up thereby. This is the cause of the splitting of drills (Fig. 4). In explanation of this splitting tendency, Fig. 5 represents a section of brittle wire *A* held in vise jaws *B*. If sufficient pressure is applied, as indicated by arrows *X* and *Y*, the wire will break at *R* and *S*, due to the resistance

of the vise and to the forces X and Y tending to revolve it about its axis.

13 Those portions of the wire not within the vise jaws represent, in the drill, the cutting edges A and B, Fig. 1, free cutting, due to their favorable lip angle. That portion of the wire held within the jaws represents the cutting edges C and D, or the chisel point, imbedded in the work and having to overcome a high cutting resistance due to the unfavorable cutting lip angle of these edges. Under very heavy feeds, these two edges tend to stand still, while edges A and B continue to cut with the result that the drill is fractured in a manner similar to the breaking of the wire in Fig. 5. In the drill the width of the vise jaws becomes infinitesimal and the fractures R and S coincide and constitute the longitudinal splitting of the drill.

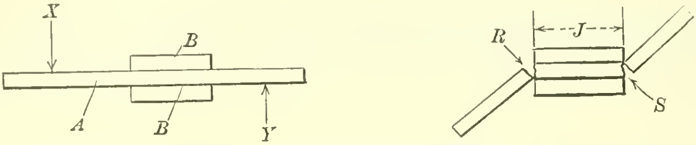

Fig. 5 Illustrating Splitting Actions by Means of Wire held in Vise

14 In the commonly used types of drills there are but two methods of reducing the chisel point, viz., by thinning the drill at its center, or by pointing. By the former method the resistance to longitudinal splitting becomes lowered and by the latter method the cutting edges lose their support at and near the center of the drill.

15 By the foregoing it will be seen that the feed possibilities of the ordinary types of drills are not very great. Either the drill will split or its cutting edges will break under heavy feeds, long before its torsional capacity is reached. Summed up, the disadvantages of this type of drill are as follows:

a Its weakness.
b The unfavorable cutting lip angles of the chisel point create a tendency to longitudinal splitting of the drills.
c If the chisel point is reduced by central thinning of the drill, the resistance to longitudinal splitting is lowered.
d If the chisel point is reduced by pointing, the cutting

edges are deprived of their essential support at and near the center of the drill.

e Essential central thickness and cutting edge support are obtained only in connection with a considerable chisel point.

f The tendency to longitudinal splitting is increased by any increase in the extent of the chisel point.

g The resistance to penetration due to the chisel point is very great and is increased by any increase in the extent of the chisel point.

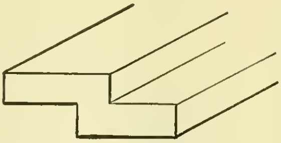

Fig. 6 Diagram illustrating Principle of Vauclain Drill

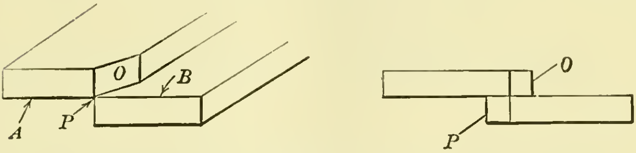

Fig. 7 Central Edges of Fig. 6 shown beveled

h It is not adapted to heavy feeds, and must therefore be used under the conditions of moderate feeds and high speeds, the least economical method of chip production.

i There are four cutting edges, two of which do not pass through the axis of motion, hence the aggregate length of the cutting edges is excessive. This is productive of greater torque, hence of power consumption.

DESCRIPTION OF THE VAUCLAIN DRILL

16 The Vauclain drill is a "heavy feed" drill adapted to the most economical method of metal cutting and of great strength, due to its design.

17 In order better to describe the drill, the section scheme will first be explained diagrammatically and the development of the actual section will then be illustrated.

18 In Fig. 6 is shown a bar comprising two flat bars overlapping and integrally connected, as indicated.

191 By beveling, as shown at *O* and *P*, Fig. 7, the edges *A* and *B* are made to meet at the axis of the bar. Since these beveled surfaces are at an angle with the axis the integral connection between the two flat bars remains unbroken.

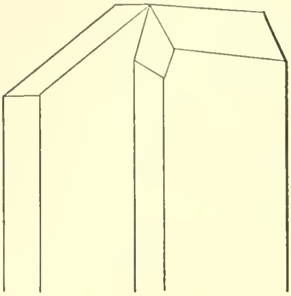

FIG. 8 SHOWING FIG. 7 GROUND TO FORM A DRILL

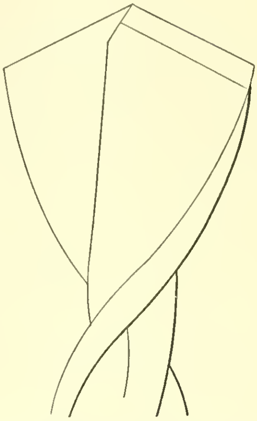

FIG. 9 SHOWING APPEARANCE OF DRILL WHEN TWISTED

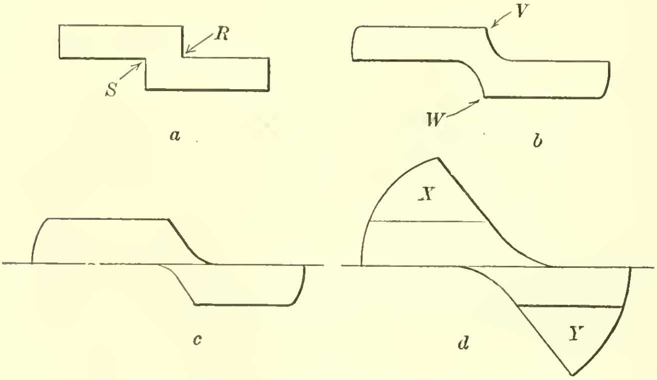

FIG. 10 SUCCESSIVE SECTIONS SHOWING DEVELOPMENT OF BAR OF FIG. 6 INTO FORM ADAPTED FOR VAUCLAIN DRILL

20 Fig. 8 indicates how the end of the bar would appear when ground to form a drill and Fig. 9 how it would appear if twisted.

21 Fig. 10-*a* again shows the section or end view of the bar in its original form. Introducing fillets at *R* and *S* gives the form shown in Fig. 10-*b* and removing the corners *W* and *V* modifies it as in Fig. 10-*c*. In Fig. 10-*d*, by adding areas *X* and *Y* to the section, its torsional value is increased.

22 Fig. 11 shows the exact section profile and in Fig. 12 is reproduced a photograph of three of these drills.

23 Tests have been made for the purpose of comparing the ordinary and the Vauclain types of drills. In order to secure uniform conditions care was taken that the different drill tests should be alike in quality of drill steel, heat treatment and tempering. The material drilled was a tough forging grade steel of about 0.45 carbon. Tables 1, 2 and 3 show the horsepower saved by giving the preference to the feeds, also the relative strength and endurance of the ordinary and the Vauclain types of drills. The size of drill used in all tests was $1^{9}/_{64}$ inches.

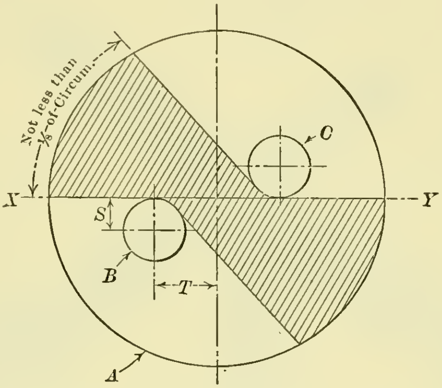

FIG. 11 DIAGRAM SHOWING EXACT FORM OF CONSTRUCTION OF DRILL

Circle *A* = Circumference of Drill
Diameter of Circle *B* = 3/16 Diameter of Drill
Diameter of Circle *C* = 3/16 Diameter of Drill
S = 3/32 of Diameter of Drill
T = 3/16 of Diameter of Drill
XY = indicates cutting edges

24 By examination of these diagrams and photographs it will be seen that the chisel point is eliminated without central thinning of the drill and without weakening the cutting edges.

ADVANTAGES OF THE VAUCLAIN DRILL

25 The improved design of this drill has brought about the following important features:

a Essential central thickness is secured.

b Cutting edges are properly supported at all points.

c Chisel point, with its unfavorable lip angles, is eliminated without weakening drill or cutting edges.

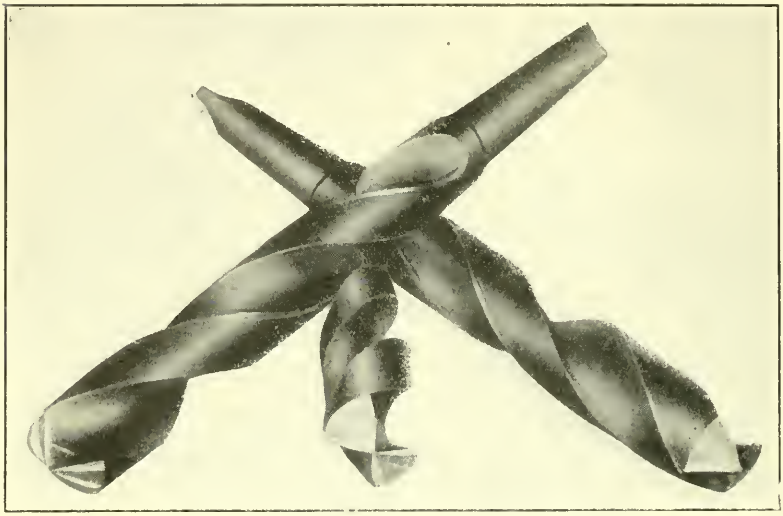

Fig. 12 Examples of the Vauclain Drill

d Cutting edges pass through axis of motion.

e There are only two cutting edges.

f Aggregate length of cutting edges is reduced to minimum.

26 As a result of these features the following advantages are secured:

g Increased strength and reduced liability to splitting.

h It is adapted to heavy feeds, the most economical method of cutting metal.

i Productive capacity is increased.

j Horsepower consumption per unit of metal removed is reduced.

k Life of drill is increased and frequency of grinding is less.

TABLE 1 STRENGTH COMPARISONS

Test	Kind of Drill	Feed per Rev.	Speed R.p.m.	Inches per Min.	Holes Drilled	Total Inches Drilled	Compare to Test	Condition of Drill After Test	See Note
1	Regular	0.009	300	2.70	1	3.875	6	O. K.	*A*
2	Regular	0.0145	300	4.35	1	3.875	7	O. K.	
3	Regular	0.020	300	6.00	1	3.875	8	O. K.	
4	Regular	0.030	300	9.00	1	3.875	9	O. K.	
5	Regular	0.050	300	15.00	0	0. –	10	Broke—see note	
6	Vauclain	0.009	300	2.70	1	3.875	1	O. K.	*B*
7	Vauclain	0.0145	300	4.35	1	3.875	2	O. K.	
8	Vauclain	0.020	300	6.00	1	3.875	3	O. K.	
9	Vauclain	0.030	300	9.00	1	3.875	4	O. K.	
10	Vauclain	0.050	300	15.00	1	3.875	5	O. K.—see note	
11	Regular	0.009	170	1.53	1	3.875	16	O. K.	*C*
12	Regular	0.0145	170	2.465	1	3.875	17	O. K.	
13	Regular	0.020	170	3.40	1	3.875	18	O. K.	
14	Regular	0.030	170	5.10	1	3.875	19	O. K.	
15	Regular	0.050	170	8.50	see note	1.000	20 note	Broke—see note	
16	Vauclain	0.009	170	1.53	1	3.875	11	O. K.	*D*
17	Vauclain	0.145	170	2.465	1	3.875	12	O. K.	
18	Vauclain	0.020	170	3.40	1	3.875	13	O. K.	
19	Vauclain	0.030	170	5.10	1	3.875	14	O. K.	
20	Vauclain	0.050	170	8.50	1	3.875	15	O. K.—see note	
21	Regular	0.009	170	1.53	1	3.875	26	O. K.	*E*
22	Regular	0.145	170	2.465	1	3.875	27	O. K.	
23	Regular	0.020	170	3.40	1	3.875	28	O. K.	
24	Regular	0.030	170	5.10	. .	3.875	29	Broke emerging	
25	Regular	0.050	170	8.50	. .	. . . . .	30-31-32	. . . . . .	
26	Vauclain	0.009	170	1.53	1	see note	21	O. K.	*F*
27	Vauclain	0.0145	170	2.465	1		22	O. K.	
28	Vauclain	0.020	170	3.40	1		23	O. K.	
29	Vauclain	0.030	170	5.10	1		24	O. K.	
30	Vauclain	0.050	170	8.50	1		25	O. K.	
31	Vauclain	0.050	208	10.40	1		25	O. K.	
32	Vauclain	0.050	245	12.25	see note	1.000	25	Broke—see note	

A. Drill broke after point had penetrated to ⅞ in. diameter, broke into many pieces; split. Torque is practically proportional to the square of the diameter, hence, since *A* broke when point had penetrated to only ⅞ in. diameter, *B* showed 170 per cent the strength of *A*.

B. Drill was permitted to emerge from work without knocking out feed; a slight crack in one cutting edge developed after drilling at rate of 15 in. per min. and emerging.

C. Drill broke into many pieces after penetrating 1 in. in last hole. Drilling at rate of 8.50 in. per min. split.

D. Since no heavier feeds were obtainable in the machine this drill was afterward run at 208 r.p.m. and 0.050 feed and broke only when emerging from bottom of hole; drilling at rate of 10.40 in. per min. and emerging. Taking into consideration that *D* emerged and *C* did not, the difference in strength is much greater than that indicated by the difference in inches per minute.

F. In order to get an actual comparison with *C* which broke at 1 inch penetration, this drill was not allowed to emerge from the work. It broke drilling at rate of 12.25 in. per min.; after penetrating 1 in. Drill broke torsionally and did not split.

l Distress in machine is less and resistance to penetration is reduced.

m Better holes are produced and afterwork of reaming is less.

TABLE 2 ENDURANCE TESTS

Test	Kind of Drill	Feed per Rev.	Speed R.p.m.	Inches per Min.	Holes Drilled	Total Inches Drilled	Compare to Test	Condition of Drill After Test	See Note
33	Regular	0.009	300	2.70	42	162.75	34	Cutting edges worn	*A*
34	Vauclain	0.020	208	4.16	42	162.75	33	Cutting edges worn	
35	Regular	0.020	300	6.00	21	81.37	36	Cutting edges burned	—*B*
36	Vauclain	0.030	208	6.24	21	81.37	35	Cutting edges O.K.	
37	Regular	0.020	300	6.00	6	23.25	38–39	Badly burned	
38	Vauclain	0.030	208	6.24	26	105.75	37	Cutting edges O.K.	*C*
39	Vauclain	0.030	208	6.24	35	135.12	37	See note	*D*
40	Vauclain	0.0145	208	3.02	20	...	41	O. K.	
41	Regular	0.009	208	1.87	20	...	40	O. K.	
42	Regular	0.009	300	2.70	11	...	43	O. K.	
43	Vauclain	0.0145	245	3.55	11	...	42	O. K.	
44	Regular	0.009	300	2.70	10	...	45	O. K.	
45	Vauclain	0.0145	245	3.55	10	...	44	O. K.	
46	Vauclain	0.020	245	4.90	5	...	47	O. K.	
47	Regular	0.0145	245	3.55	5	...	46	O. K.	
48	Regular	0.0145	300	4.35	10	...	49–52	O. K.	
49	Vauclain	0.020	300	6.00	10	...	48	O. K.	
50	Regular	0.020	300	6.00	25	...	51	Burned corners	*B*
51	Vauclain	0.020	300	6.00	39	...	50	Burned corners	*E*
52	Vauclain	0.020	300	6.00	10	...	48	O. K.	

A Cutting edges of Vauclain drill *less* worn than regular drill.

B Chisel point also burned.

C Broke in 26th hole by running into another hole.

D Spindle of machine ran hot, causing "jumping" of speed, and cutting edges; broken by resulting shocks. Drill at one period stood still, but did not fracture except at cutting edges.

E Point O. K.

TABLE 3 COMPARATIVE TESTS OF REGULAR AND VAUCLAIN DRILLS

Test	Kind	Feed per Rev.	Speed, R.P.M.	In. per Min.	H.P. Expended	Per Cent Saving
53	Regular	0.00599	300	1.797	10.86	....
54	Regular	0.02	87	1.74	10.30	....
55	Vauclain	0.02	87	1.74	5.03	51.2
56	Vauclain	0.00599	300	1.797	4.73	56.4

TESTS OF A 1000-H.P. TUBES HIGH B. & W. BOILER

By B. N. Bump

ABSTRACT OF PAPER

The tests described in this paper were made on an inclined header Babcock and Wilcox boiler, with superheater and mechanical stoker. The experimental unit was installed for the purpose of determining the maximum heat efficiency and fuel economy, and the variation of such efficiency with changes in the boiler rating, also to gather data useful in the design of a unit for a large installation. The peculiar feature of the boiler is the unusual height; the sections being 24 tubes high.

A Foster superheater was placed between the top row of tubes and the boiler drums, and the gases entered the superheater after passing over 24 tubes. The superheat was found to go up with the boiler rating, increasing from 30 deg. to about 75 deg. fahr. for a change in rating from 55 to 120 per cent.

The fuel was burned on a six retort Taylor stoker having a grate area, exclusive of dump grate, of 62.5 sq. ft. One of the results of increasing the height of a boiler is a large ratio of boiler heating surface to grate area, which in this case was 160 to 1. Such a large ratio limits the capacity of the boiler, making it impossible to run at high boiler ratings.

The best combined efficiency, 81.3 per cent, was obtained at about 56 per cent of rating. The efficiency decreases slowly with an increase in the boiler rating. The low rating tests show exit gas temperatures but little above the temperature of the saturated steam in the boiler. The radiation losses are low considering the exposed condition and the low atmospheric temperatures surrounding the unit.

TESTS OF A 1000-H.P. 24 TUBES HIGH B. & W. BOILER

By B. N. Bump, Syracuse, N. Y.,

Member of the Society

The boiler tested was erected as an experiment for the purpose of determining, by test and by continuous operation in the regular service of a large plant, the advantages and disadvantages of so high a boiler of the B. & W. type. These tests were made from November 1911 to February 1912. They were expected to answer the following questions:

a At what rating will the highest efficiency be obtained

b Will the superheat obtained, with the superheater located above the 24th tube, be sufficient to pay for the installation of the superheater

c Will the last pass of the boiler be effective

d Can the exit gas temperature be reduced very nearly to the temperature of the steam in the boiler

e If the exit gas temperatures are close to the steam temperature, will pitting occur in the back end of the boiler due to the sulphur content of the gas

BOILER AND SUPERHEATER

2 The boiler was built up of one regular 14-tube section and above this, with a space of some 11 in. between them, was placed a 10-tube section. The headers of the 10-tube and 14-tube sections were joined by short nipples, making a 24 high header. There are 21 of these sections containing a total of 504 tubes. The tubes are 4 in. in diameter, and 18 ft. long. There are 3 drums 42 in. in diameter, and 24 ft. long. The water heating surface of the boiler is 10,000 sq. ft. Between the top tubes and

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the drums a Foster superheater is placed containing 1750 sq. ft. of superheating surface.

3 The gases make three passes through the boiler. The baffles are arranged to give a gradual decrease in space through the first and second passes in the direction of flow, which tends to maintain the velocity of the gases as the temperature decreases. The gases enter the superheater from the top of the first pass, after traveling over 24 tubes. From the superheater they turn downward through the second pass; upward through the third pass, and then make their exit between the downtakes, and through the rear wall to the stack. The spaces between the header, both front and rear, were packed with asbestos.

STOKER AND FURNACE

4 The boiler is fired by a six-retort Taylor stoker, grate area 62.465 sq. ft., not including any dump grate area. The ratio of boiler heating surface to grate area is 160 to 1. The ratio of boiler plus superheater surface to grate area is 188 to 1.

5 The height of the combustion chamber is about 6.5 ft., and the width of the furnace 12 ft. 7 in., which latter is greater than the width of the stoker to allow for placing a cast-iron wind box on either side of the stoker. This is believed to be a new feature, tried as an experiment. Previously there had been trouble from the formation of clinker on the side walls with stokers of both the underfeed and overfeed type, owing to some coals giving a tough sticky clinker which is very troublesome. With the side walls against the stoker, clinker forms rapidly, and is difficult to remove. Setting back the side walls lessens clinker formation to some extent, and facilitates the removal of the clinker.

6 A brick ledge between the stoker and side wall is objectionable, because in time clinker will fasten to it and it is almost impossible to get a bar between clinker and ledge. The cast-iron wind boxes overcome the objections to the brick ledge by chilling the clinker so that it does not stick. A bar can always be forced between the clinker and wind box to raise the clinker. A part or all of the air from the blast fan is passed through these wind boxes on its way into the stoker. After six months of continuous service there is no sign of burning on the wind boxes.

7 Fig. 1 shows a front and side elevation of the unit. On the side elevation the locations of the pyrometer rods and gas sampling tubes are shown. The furnace temperatures were taken

through the three openings shown just under the lower tubes. The other gas temperatures were taken at positions 1 to 7 inclusive. The numbers on the data sheets and charts of gas temperatures refer to these positions.

8 The walls were lined with 9 in. of fire brick backed up by 9 in. of porous insulating brick. The outer surfaces of all walls were painted with two coats of boiler pitch to prevent any filtration of air through the brick work. Against these painted surfaces of the front and side walls 85 per cent magnesia covering 4 in. thick was placed. The magnesia was held against the brick and protected by a thin steel shell. Asbestos board was placed on the top half of the drums, then loose mineral wool was placed over the entire top of the boiler.

TEST APPARATUS AND DATA

9 The tests varied from 18 to 48 hours in length, although most of them were about 24 hours long. The coal was weighed in a wheelbarrow on scales tested frequently for accuracy. A small sample of coal was taken from each wheelbarrow load to make up the test sample for moisture, analysis, and calorific value.

10 The feedwater was measured in a tank of 13,000 gal. capacity. All feedwater piping was under observation, and any leaks were quickly discovered. The drips from the pump plungers were caught and returned to the feed tank. The feedwater piping being entirely independent there was no possibility of the measured water going anywhere except into the boiler. The blow-off piping was slip blanked, and the boiler blown down only between tests. The boiler was examined for leaks at frequent intervals, and the boiler tubes were dusted every 48 hours in the intervals between tests.

11 Steam pressures, superheated steam temperatures, and steam calorimeter temperatures were taken by recording instruments.

12 It was found necessary to have some suction over the fire to prevent the escape of gas into the boiler room. This suction was kept as low as possible. Gas temperatures were taken at seven different points as given in Table 5. The seven pyrometer rods were wired to one indicating instrument so that, by the use of knife edge switches, the indication of one rod after another could be taken in quick succession.

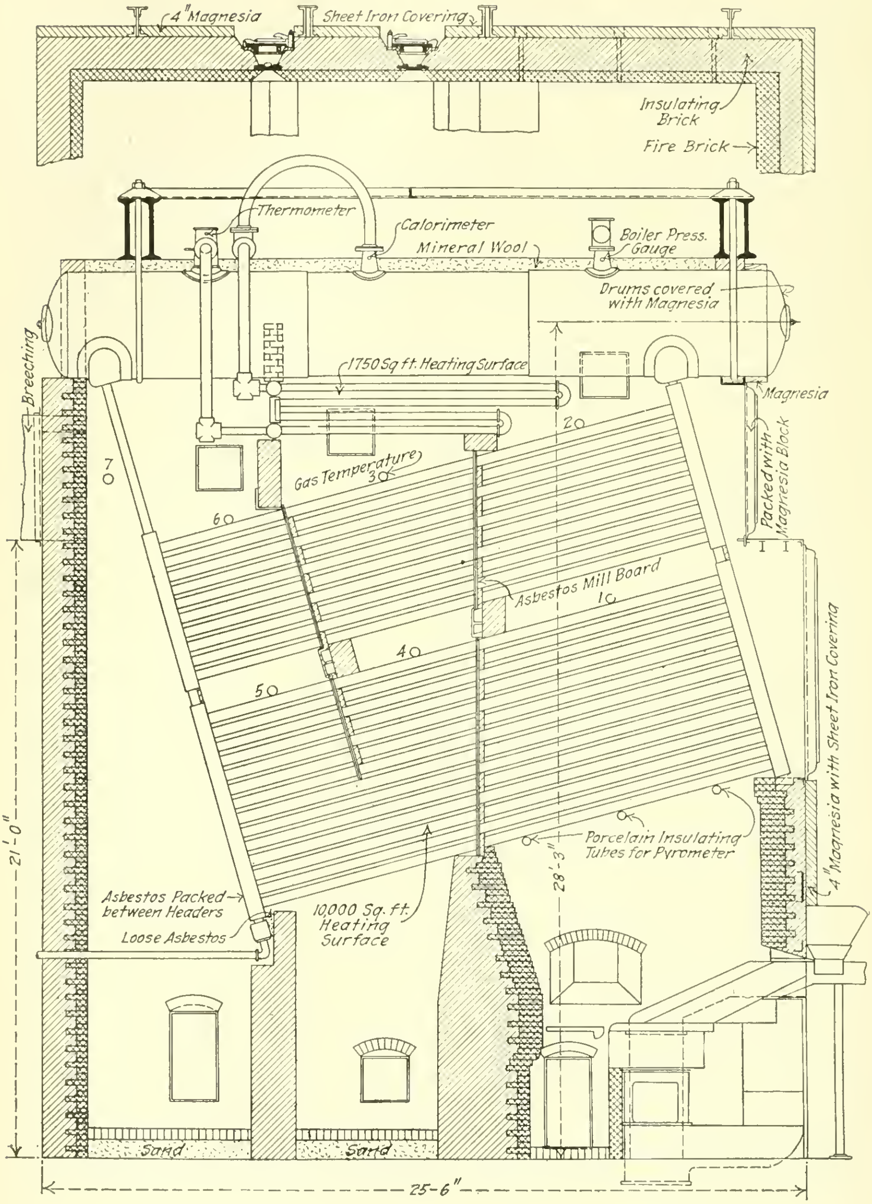

4" Magnesia
Sheet Iron Covering
Insulating Brick
Fire Brick
Thermometer
Calorimeter
Mineral Wool
Boiler Press. Gauge
Drums covered with Magnesia
Breeching
1750 Sq ft. Heating Surface
Magnesia
Packed with Magnesia Block
2
7
Gas Temperature
3
6
Asbestos Mill Board
1
4
5
4" Magnesia with Sheet Iron Covering
Porcelain Insulating Tubes for Pyrometer
21'-0"
28'-3"
Asbestos Packed between Headers
Loose Asbestos
10,000 Sq. ft. Heating Surface
Sand
Sand
25-6"

Fig. 1-a Longitudinal Section of Boiler

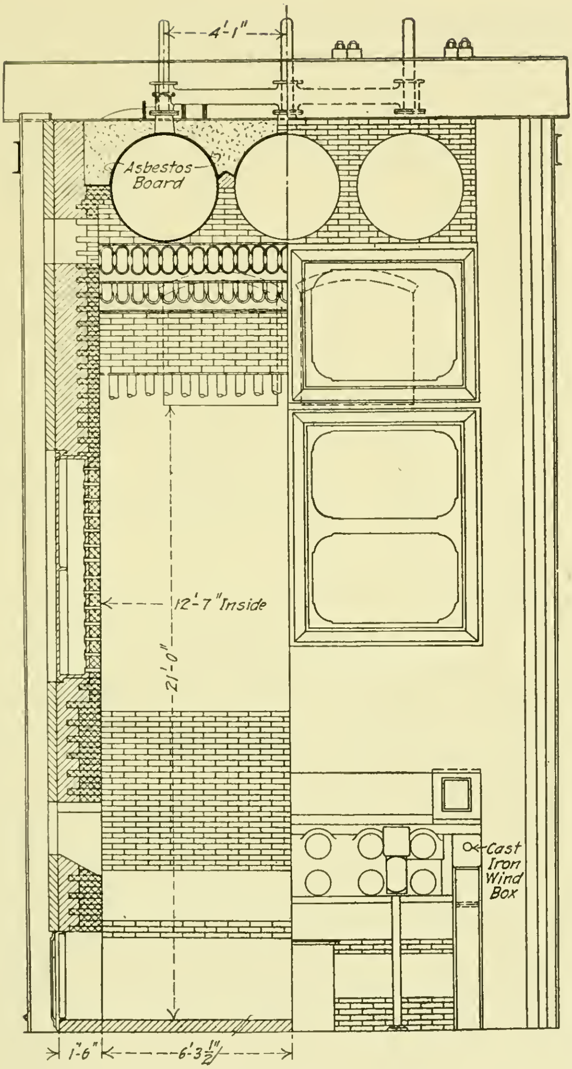


FIG. 1-b FRONT ELEVATION OF BOILER

13 A high reading pyrometer was used for furnace temperatures. There was considerable trouble in keeping this pyrometer in working order and consequently the furnace temperatures are given for only a few of the tests.

14 Samples of gas for analysis were taken in the first pass at the top of the 14th tube, and before the breeching. The samples were taken through six sampling tubes placed in the space between the boiler and back wall, and directly in front of the opening to the breeching. One tube was connected to a CO_2 recorder and the other five were attached to large sampling bottles which were run continuously. The tips of the tubes were so placed that the samples taken through each represented approximately equal volumes passing to the stack.

15 The first 19 tests were made with mixed coal, from four different mines all in the same region. Tests Nos. 21 to 40 were all run on coal from No. 29 mine, lighter than that used on the early tests and giving much less clinker trouble. It made a loose fuel bed and allowed more boiler capacity with less draft in the stoker. For best combined efficiency the dry coal per square foot of grate per hour is about 26 lb., or nearly 270 lb. per retort. The quantity of coal burned per square foot of grate per hour was increased to nearly 63 lb. with a decrease in combined efficiency of 5.5 per cent.

16 To determine the carbon in the refuse the total refuse was spread out upon a large floor, broken up, thoroughly mixed and quartered. The quarter sample was ground, and this ground portion was in turn quartered to get the sample for chemical analysis. The calorific values of the coals were determined by the oxygen-bomb calorimeter.

17 The radiation and unaccounted for losses have been calculated for those tests, for which there were ultimate analyses of the coal. The tests were made during extremely cold weather, and there was over the boiler only a temporary building affording but little protection from the elements; the boiler room temperatures were so low at times the boiler pressure gage piping and the water column blow-off piping froze. In view of these facts and the drafty condition of the boiler room the radiation and unaccounted for losses of from 5 to 7 per cent are worthy of note.

18 The theoretical efficiency was calculated on the assumption that the only heat loss was that to the stack when the combus-

tion was complete without excess air, and the temperature of gas leaving the boiler was equal to the temperature of the wet steam in the boiler. The efficiencies in Table 1 have been calculated for those tests for which there were ultimate analyses of the coal.

19 It may be possible under favorable conditions to reduce the radiation losses to 2.5 per cent. With thorough mixing and a suitable type of combustion chamber the heat losses in the gas may be expected to be not more than 1.5 per cent greater than the theoretical gas loss; the maximum combined efficiency which may be attained under these most favorable conditions is about 85 per cent.

TABLE 1 THEORETICAL EFFICIENCIES

No. of Tests	Theoretical Efficiency, Per Cent	Combined Efficiency, Per Cent	Gain in Efficiency necessary to attain Theoretical Efficiency
8	90.10	79.60	10.50
11	90.15	78.14	12.01
16	90.05	76.18	13.87
18	89.95	75.20	14.75
25	89.52	75.79	13.73
35	89.90	78.33	11.57
39	89.85	78.65	11.20

SUMMARY OF RESULTS

20 The best combined efficiencies are obtained with 56 to 66 per cent of the boiler rating. The efficiencies fall off slowly as the quantity of steam generated is increased. The extreme variation in efficiency shown by the individual tests is from 75.2 to 81.3 per cent.

21 The gain in efficiency due to the superheater seems to bear no definite relation to the amount of moisture in the steam entering the superheater. By gain in efficiency due to superheater is meant the heat absorbed by the superheater in percentage of the total heat in the coal. The superheat shows a general tendency upward with the increase in boiler rating. For 60 per cent of the boiler rating the superheat is about 36 deg. fahr., and for 120 per cent rating about 74 deg. fahr. At 60 per cent rating the weight of steam passing through the super-

heat is 18,000 lb. per hour, the amount for which the superheater was built. The pressure drop through the superheater for its rated quantity of steam is 5 lb.; at 120 per cent boiler rating the weight of steam passing through the superheater is double that for which the superheater was designed, and the pressure drop through superheater is 28 lb.

22 At the low boiler ratings the last pass of the boiler, and in fact the last two passes, are of very little use. The best combined efficiencies were obtained when running between 56 and 66 per cent of boiler rating. At these low ratings the drop in gas temperature through the second and third passes altogether is about 20 deg. fahr., and the heat absorbed is very small, about 1.3 per cent of the total. These two passes have 50 per cent of the boiler heating surface. If the second and third passes were dropped off entirely, the loss in combined efficiency would be about 1 per cent at the low rating. It is only when the boiler has reached 75 per cent rating or more that the gain in economy in the last two passes is sufficient to give a reasonable return upon the investment in heating surface. While the exit gas temperatures were reduced almost to the lowest theoretical limit which can be reached without the use of an economizer, it was done at a large expenditure in heating surface.

23 When running at about 50 per cent of boiler rating the temperature of the gases leaving the boiler is practically that of the steam in the boiler. As the capacity is increased the difference between the temperature of the gases leaving the boiler and the temperature of the steam in the boiler increases. The increase in the difference between these two temperatures seems to be approximately in the same ratio as the increase in capacity, so that for an increase from 50 to 100 per cent of the boiler rating the difference in temperature between exit gases and steam increases about 50 deg. fahr.

24 An examination of the boiler after more than six months' service showed no evidence of pitting of the heating surface of the last pass due to the sulphur content of the coal and low exit gas temperature.

APPENDIX

RATING VS. COMBINED EFFICIENCY

25 Fig. 2, showing the relation between boiler rating and combined efficiency, indicates that the efficiency decreases slowly as the steam generated is increased. The line shows an efficiency of 80 per cent at 50 per cent of rating, and an efficiency of 73.5 per cent at 150 per cent rating, or a decrease of 6.5 per cent in the combined efficiency for an increase of 100 per cent in rating. This decrease in efficiency with an increase in rating seems to be due primarily to the temperature of the gases leaving the boiler.

EFFICIENCY AND RATING VS. TEMPERATURE OF GASES LEAVING BOILER

26 It may be seen from Fig. 2 that the temperature of the gases leaving the boiler goes up with the rating; the increase being about 1 deg. fahr. for 1 per cent rating. On the other hand an increase in the exit gas temperature gives a decrease in the efficiency. As an increase in exit gas temperature gives an increased rating and a decreased efficiency it follows that the efficiency will fall off as the rating increases.

27 The superheat-rating curve shows that the superheat has a general tendency upward with the rating.

28 The line for rating coal per retort per hour shows that the coal consumption increases at a more rapid rate than the steam generated.

GAS TEMPERATURES

29 The gas temperatures throughout the boiler from furnace to breeching, and the percentages of total heating surface (boiler plus superheater) between the several pyrometer positions are given in the tables, and have been plotted also. While these percentages vary irregularly, and seem decidedly freakish in individual cases, the general tendency is to show a decrease in the percentage of heat absorption in the first pass with an increase in boiler rating. In the second and third passes the opposite condition prevails, the percentage of heat absorption increasing with boiler rating.

30 A study of the gas temperature data will explain these conditions. At the low boiler rating the gas temperatures are so low by the time the gases reach the second pass that the rate of absorption is low and the percentage small. In general the temperature of the gas entering the second pass increases with the boiler rating; this increases the rate of heat absorption, and gives to this portion of the boiler a greater proportion of the total work.

TABLE 2 1000-H.P. BOILER, RESUMÉ OF PRINCIPAL RESULTS

Test No.	Per Cent Rating	Combined Efficiency	Temperature of Gases leaving Boiler, Deg. Fahr.	CO_2 in Gases leaving Boiler, Per Cent	Per Cent Combustible in Dry Refuse	B.t.u. per Lb. Dry Coal	Evaporation from and at 212 Deg. Fahr. per Lb. Dry Coal Combined Unit	Quality of Steam leaving Boiler	Radiation and Unaccounted for Losses
1	84.79	78.02	390	10.32	15.49	14334	11.525	97.92	....
3	56.72	80.89	363	.....	10.30	14541	12.121	97.83	....
4	61.77	77.93	368	10.10	10.09	14517	11.658	97.93	....
5	59.48	79.10	361	9.39	13.91	14498	11.818	97.99	....
6	61.43	80.76	368	9.00	8.64	14499	12.067	98.16	....
7	60.87	81.00	367	9.00	1.44	14526	12.125	98.20	....
8	60.60	79.60	367	10.38	3.90	14493	11.889	98.11	5.46
9	57.36	77.71	364	10.16	3.39	14445	11.567	98.00	....
10	55.88	81.30	362	10.02	9.69	14289	11.972	97.97	....
11	67.94	78.14	369	10.30	10.93	14624	11.775	97.94	6.42
12	66.43	80.69	373	11.20	17.91	14454	12.018	98.11	....
13	68.86	78.89	375	10.50	18.08	14396	11.704	98.01	....
14	74.25	78.64	381	10.59	14.75	14602	11.833	98.23	....
15	84.94	76.88	390	10.44	23.79	14467	11.461	98.15	....
16	76.21	76.18	383	10.54	23.52	14590	11.453	98.14	6.99
17	76.77	78.76	383	10.68	21.10	14618	11.864	98.20	....
18	89.08	75.20	401	10.22	20.05	14645	11.349	98.00	7.15
19	90.75	75.42	396	10.77	24.50	14471	11.247	98.01	....
21	108.83	76.21	412	10.75	28.73	14518	11.401	98.03	....
22	106.90	75.50	415	10.89	21.48	14462	11.252	98.02	....
23	93.93	76.18	399	10.15	17.80	14263	11.197	97.84	....
24	107.66	76.87	412	10.00	25.76	14522	11.504	98.00	....
25	127.70	75.79	423	11.66	30.68	14415	11.258	97.99	5.65
26	121.36	75.93	426	11.35	29.18	14290	11.181	98.42	....
27	75.65	77.28	379	10.17	14.94	14284	11.375	98.05	....
29	71.63	78.76	367	10.01	14.37	14254	11.569	98.10	....
30	73.75	77.49	369	9.84	16.23	14430	11.523	98.11	....
33	77.80	76.48	376	9.24	20.14	14380	11.333	98.09	....
34	67.53	78.77	374	9.31	7.99	14376	11.670	98.05	....
35	63.05	78.33	376	9.39	7.07	14455	11.668	98.04	5.00
37	62.46	79.08	369	9.42	7.59	14409	11.742	98.10	....
39	53.98	78.65	360	8.84	4.69	14515	11.764	97.80	4.75
40	53.72	78.79	360	9.43	5.09	14378	11.675	97.93	....

Tests 1–19 inclusive with mixed coal from mines Nos. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

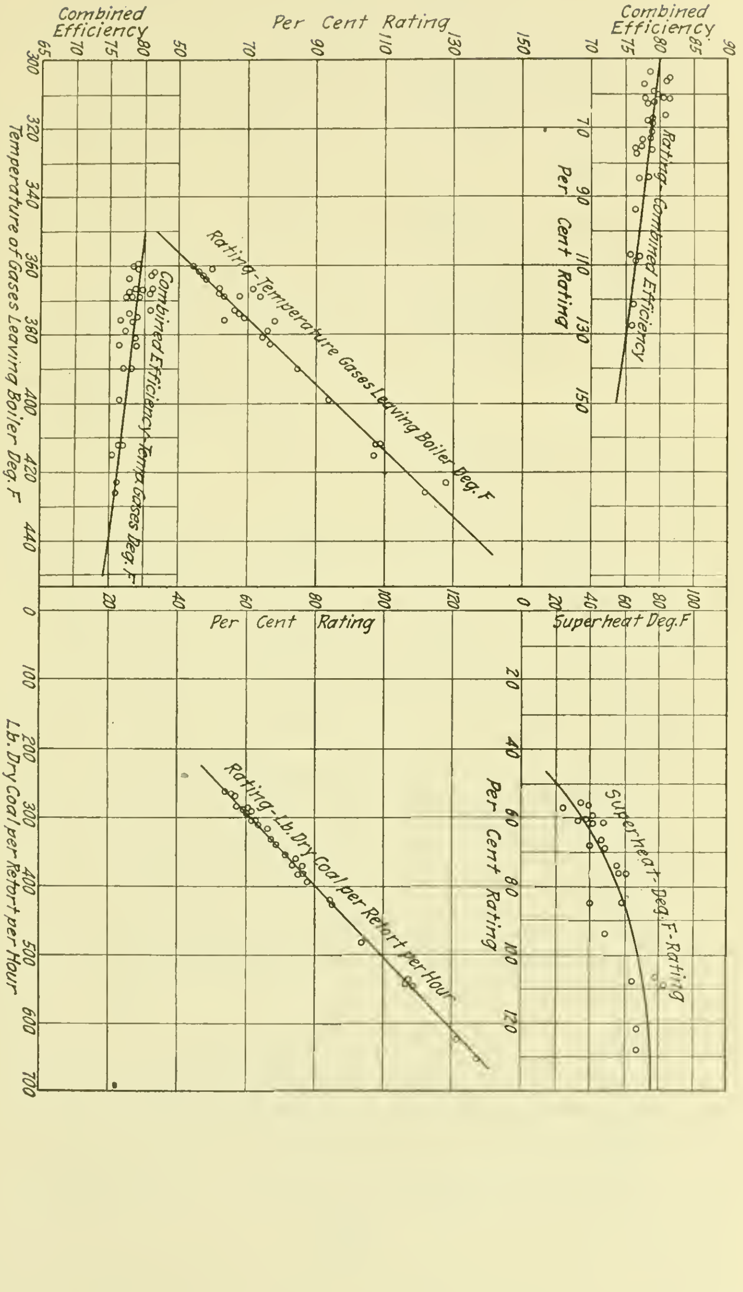


FIG. 2 CURVES SHOWING RELATION BETWEEN BOILER RATING AND COMBINED EFFICIENCY

TABLE 3 1000-H.P. BOILER, TEST DATA

Test No.	Date of Test	Length of Test, Hours	Steam Pressure by Gage Lb. per Sq. In.		Barometer, Lb.	Average Draft, In. of Water			Temperature, Deg. Fahr						
			In Boiler	Leaving Superheater		Pressure in Stoker Wind Box	Suction over Fire	Suction before Breeching	Fireroom	Saturated Steam at Pressure, leaving Superheater	Steam leaving Superheater	Superheat	Feedwater entering Boiler	Gases leaving Boiler	Saturated Steam at Boiler Pressure
1	11-16-17-18-11	47.983	136.00	124.00	14.53	4.89	0.157	.....	41.9	352.26	392.00	39.74	147.92	390	358.77
3	11-21-22-11	24.28	127.00	124.00	14.57	2.480	0.148	.....	32.0	352.29	392.00	39.71	133.34	363	353.95
4	11-23-24-11	24.61	130.00	124.00	14.58	2.599	0.144	.....	44.6	352.27	393.80	41.53	142.34	368	355.59
5	11-24-25-11	21.61	126.00	121.00	14.57	2.599	0.146	.....	32.0	350.58	392.00	41.42	141.26	361	353.39
6	11-27-28-11	17.90	131.00	125 00	14.61	2.520	0.137	.....	48.2	352.85	401.00	48.15	146.12	368	356.11
7	11-28-29-11	21.50	130.00	125.00	14.54	2.599	0.137	.....	47.75	352.81	390.20	37.39	144.68	367	355.57
8	12-1-2-11	28.083	131.00	126.00	13.51	2.717	0.104	.....	38.66	354.81	387.50	32.69	142.16	367	355.55
9	12-4-5-11	24.33	126.00	123.00	14.75	2.402	0.098	.....	39.56	351.84	375.80	23.96	125.24	364	353.48
10	12-5-6-11	24.68	128.00	124.00	14.74	2.402	0.137	.....	40.64	352.36	386.60	34.24	125.60	362	354.58
11	12-6-7-11	21.66	141.00	128.00	14.74	3.228	0.126	.....	45.50	354.56	393.80	39.24	131.72	369	361.44
12	12-7-8-11	20.76	136.00	128 00	14.77	3.386	0.122	.....	46.04	354.58	401.00	46.42	129.74	373	358.90
13	12-8-9-11	24.30	136.00	127.00	14.89	3.662	0.137	.....	46.22	354.13	402.80	48.67	125.24	375	358.95
14	12-11-12-11	24.38	135.00	124.00	14.69	3.465	0.110	.....	59.72	352.33	408.20	55.87	145.04	381	358.30
15	12-12-13-11	23.76	134.00	126.00	14.63	4.370	0.118	.....	43.16	353.41	411.80	58.39	139.64	390	357.73

TABLE 3—*Continued*

16	12–13–14–11	18.68	135.00	129.00	14.71	3.110	0.098	.....	39.56	355.12	411.80	56.68	137.48	383	358.32
17	12–14–15–11	23.93	134.00	127.00	14.69	3.465	0.091	0.196	40.10	354.00	415.40	61.40	157.46	383	357.42
18	12–19–20–11	29.11	137.00	126.00	14.84	4.567	0.118	0.359	34.88	353.52	410.00	56.48	137.66	401	359.42
19	12–21–22–11	24.11	140.00	127.00	14.72	5.276	0.137	0.394	36.50	354.02	408.20	54.18	130.64	396	360.86
21	12–26–27–11	22.13	147.00	130.00	14.30	5.040	0.085	0.472	43.88	355.44	438.80	83.36	131.18	412	364.24
22	12–27–28–11	23.70	143.60	127.00	14.42	4.764	0.098	0.484	36.50	353.84	433.40	79.56	119.84	415	362.70
23	12–28–29–11	20.70	144.00	127.00	14.65	4.685	0.173	0.496	40.64	353.98	402.80	48.82	125.42	399	362.93
24	12–29–30–11	23.55	145.00	128.00	14.78	4.488	0.122	0.500	37.94	354.58	419.00	64.42	145.22	412	363.49
25	1–4–5–12	18.00	158.00	129.00	14.53	5.630	0.085	0.461	25.16	355.01	422.60	67.59	141.80	423	369.66
26	1–5–6–12	23.96	150.00	123.00	14.61	5.906	0.075	0.512	12.20	351.77	419.00	67.23	140.90	426	365.85
27	1–18–19–12	22.28	141.00	134.00	14.56	3.071	0.158	0.276	43.70	357.67	404.60	46.93	134.06	379	361.35
29	1–22–23–12	23.95	138.00	129.00	14.57	3.110	0.110	0.173	29.30	355.05	388.40	33.35	139.10	367	359.80
30	1–23–24–12	22.36	137.00	129.00	14.49	3.189	0.106	0.268	31.64	354.99	395.60	40.61	139.10	369	359.26
33	1–26–27–12	24.15	139.00	131.00	14.58	3.583	0.126	0.389	12.56	356.09	392.00	35.91	138.92	376	360.28
34	1–29–30–12	24.016	138.00	132.00	14.53	2.717	0.110	0.358	29.48	356.61	392.00	35.39	146.30	374	359.76
35	1–30–31–12	20.40	134.00	128.50	14.46	2.638	0.126	0.354	29.30	354.69	381.20	26.51	157.28	376	357.64
37	2–1–2–12	25.26	134.00	128.00	14.56	2.795	0.110	0.252	22.46	354.47	377.60	23.13	152.24	369	357.67
39	2–6–7–12	24.53	132.00	127.00	14.60	2.126	0.079	0.126	26.24	353.95	377.60	23.65	157.28	360	356.67
40	2–7–8–12	18.28	135.00	130.00	14.55	2.205	0.067	0.078	26.42	355.57	368.60	13.03	160.70	360	358.23

Tests 1–19 inclusive with mixed coals from mines Nos. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

TABLE 4 100-H.P. BOILER, TEST RESULTS

Test No.	Relative Humidity of Atmosphere, per Cent	Feedwater, Lb. per Hr.	Evaporation Factor Combined Unit	Evaporation Factor Boiler only	Lb. Dry Coal per Hr.	Lb. Dry Refuse per Hr.	Lb. Combustible ascending from Grate per Hr.	Lb. Dry Coal per Hr. per Sq. Ft. Grate.	Lb. Dry Coal per Hr. per Retort	Equivalent Evaporation per Lb. Dry Coal Combined Unit	Equivalent Evaporation per Lb. Dry Coal Boiler only	Per Cent Builders Rating Developed	Efficiencies			Gain in Efficiency Due Superheater
													Boiler, Furnace, and Superheater	Boiler and Furnace	Boiler Only	
1	....	25814	1.13317	1.09201	2538	263	2275	40.63	423.00	11.525	11.106	84.79	78.02	75.18	75.20	2.84
3	....	17045	1.14817	1.10516	1615	144	1471	25.85	269.17	12.121	11.666	56.72	80.89	77.85	77.54	3.04
4	....	18692	1.13998	1.09712	1828	150	1678	29.26	304.67	11.658	11.219	61.77	77.93	74.99	74.62	2.94
5	....	17994	1.14048	1.09829	1736	158	1579	27.79	289.33	11.818	11.381	59.48	79.10	76.18	76.25	2.92
6	....	18590	1.14004	1.09539	1756	154	1603	28.11	292.67	12.067	11.594	61.43	80.76	77.59	77.43	3.17
7	....	18487	1.13592	1.09710	1732	138	1594	27.73	288.67	12.125	11.711	60.87	81.00	78.23	77.43	2.77
8	....	18400	1.13632	1.09888	1759	143	1616	28.16	293.17	11.889	11.498	60.60	79.60	76.99	76.30	2.61
9	....	17253	1.14706	1.11486	1711	138	1573	27.39	285.17	11.567	11.243	57.36	77.71	75.53	74.71	2.18
10	....	16719	1.15298	1.114427	1610	158	1453	25.77	268.33	11.972	11.752	55.88	81.30	79.81	78.17	1.49
11	....	20366	1.15026	1.10953	1989	158	1831	31.84	331.50	11.775	11.358	67.94	78.14	75.37	75.22	2.77
12	....	19817	1.15645	1.11249	1907	184	1723	30.53	317.83	12.018	11.561	66.43	80.69	77.61	78.05	3.08
13	....	20440	1.16219	1.11622	2030	206	1824	32.50	338.30	11.704	11.241	68.86	78.89	75.77	76.23	3.12
14	....	22365	1.14537	1.09763	2165	181	1984	34.66	360.80	11.833	11.339	74.25	78.64	75.36	75.40	3.28
15	....	25424	1.15264	1.10235	2557	246	2311	40.93	426.20	11.461	10.961	84.94	76.88	73.52	74.16	3.36

TABLE 4—*Continued*

16	....	22776	1.15445	1.10462	2296	209	2087	36.76	382.66	11.453	10.958	76.21	76.18	72.88	73.47	3.30
17	....	23311	1.13620	1.08445	2233	189	2044	35.75	372.17	11.864	11.323	76.77	78.76	75.17	75.67	3.59
18	....	26640	1.15365	1.10354	2708	243	2465	43.35	451.33	11.349	10.857	89.08	75.20	71.94	72.45	3.26
19	53.0	26997	1.15974	1.11113	2784	261	2523	44.57	464.00	11.247	10.776	90.75	75.42	72.26	73.20	3.16
21	53.5	31933	1.17576	1.11153	3293	339	2954	52.72	548.80	11.401	10.778	108.83	76.21	72.04	73.50	4.17
22	49.5	31128	1.18480	1.12276	3278	333	2945	52.48	546.30	11.252	10.663	106.90	75.50	71.55	72.50	3.95
23	55.1	27889	1.16200	1.11551	2894	315	2579	46.33	482.30	11.197	10.749	93.93	76.18	73.13	73.70	3.05
24	55.3	32282	1.15060	1.09658	3229	318	2911	51.69	538.20	11.504	10.964	107.66	76.87	73.26	74.29	3.61
25	51.3	38110	1.15605	1.10142	3913	443	3470	62.64	652.20	11.258	10.726	127.70	75.79	72.21	74.02	3.58
26	53.4	36225	1.15576	1.10538	3744	452	3292	59.94	624.00	11.181	10.694	121.36	75.93	72.62	74.23	3.31
27	51.0	22631	1.15327	1.10808	2294	234	2060	36.72	382.30	11.375	10.929	75.65	77.28	74.25	74.36	3.03
29	53.0	21690	1.13932	1.10299	2136	224	1912	34.20	356.00	11.569	11.200	71.63	78.76	76.25	76.43	2.51
30	51.9	22249	1.14357	1.10299	2208	215	1993	35.35	368.00	11.523	11.114	73.75	77.49	74.74	74.94	2.75
33	51.7	23520	1.14127	1.10314	2369	237	2131	37.93	394.80	11.333	10.954	77.80	76.48	73.92	74.37	2.56
34	54.0	20553	1.13357	1.09515	1996	183	1813	31.95	332.70	11.670	11.275	67.53	78.77	76.11	75.71	2.66
35	53.0	19485	1.11639	1.08318	1864	157	1707	29.84	310.70	11.668	11.321	63.05	78.33	76.00	75.55	2.33
37	52.3	19249	1.11952	1.08892	1835	162	1673	29.38	305.80	11.742	11.421	62.46	79.08	76.92	76.61	2.16
39	49.4	16710	1.11448	1.08081	1583	124	1459	25.34	263.80	11.764	11.409	53.98	78.65	76.27	75.66	2.38
40	50.0	16775	1.10487	1.07883	1587	138	1449	25.41	264.50	11.675	11.400	53.72	78.79	76.94	76.33	1.85

Tests 1–19 inclusive with mixed coals from mines No. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

TABLE 5 1000-H.P. BOILER, TEMPERATURE AND DRAFTS

Test No.	Gas Temperatures, Deg. Fahr.								Draft in Stoker and Wind Boxes, In. Water						
	Furnace	No. 1 {In First Pass over 14th Tube}	No. 2 {In First Pass over 24th Tube}	No. 3 {In Second Pass over 24th Tube}	No. 4 {In Second Pass over 14th Tube}	No. 5 {In Third Pass over 14th Tube}	No. 6 {In Third Pass over 24th Tube}	No. 7 {Leaving Boiler between Boiler and Back Wall}	East Tuyere Box	West Tuyere Box	Bottom West Outside Box	Bottom East Outside Box	Center or Main Wind Box	Top East Outside Box	Top West Outside Box
1	....	759	572	460	423	388	385	390	4.01	4.13	4.76	5.19	4.89	6.14	5.98
3	....	691	518	387	399	378	378	363	2.16	2.20	2.87	2.95	2.48	4.09	4.01
4	....	631	498	406	361	365	363	368	2.32	2.40	2.91	3.18	2.59	4.05	3.89
5	....	639	509	406	365	367	370	361	2.32	2.40	2.91	3.11	2.59	4.05	3.97
6	....	666	500	408	390	378	367	368	2.32	2.32	2.91	3.07	2.51	4.01	3.97
7	....	639	482	387	381	370	363	367	2.36	2.36	2.95	3.11	2.59	3.97	3.93
8	1763	658	487	385	370	370	369	367	2.44	2.51	2.99	3.14	2.71	4.05	3.97
9	1841	626	417	365	352	356	363	364	2.16	2.24	2.79	2.87	2.40	3.89	3.81
10	1800	644	493	379	365	363	370	362	2.24	2.32	2.83	2.87	2.40	3.93	3.85
11	1888	696	505	387	383	363	372	369	2.95	3.07	3.50	3.66	3.22	4.60	4.52
12	1866	738	532	396	392	370	379	373	3.18	3.22	3.66	3.81	3.38	4.68	4.60
13	1958	801	572	428	414	385	390	375	3.42	3.50	3.85	4.09	3.66	4.68	4.60
14	1902	738	554	424	408	379	381	381	3.18	3.22	3.66	3.97	3.46	4.60	4.52
15	1989	838	621	477	441	396	401	390	4.05	4.17	4.76	5.07	4.37	6.10	5.94

TABLE 5—*Continued*

16	....	727	538	423	390	378	374	383	2.83	2.95	3.50	3.74	3.11	4.68	4.60
17	....	788	594	433	408	385	385	383	3.14	3.30	3.70	3.97	3.46	4.68	4.60
18	....	851	651	496	394	405	403	401	4.13	4.29	4.76	5.19	4.56	5.90	5.82
19	....	810	612	518	441	401	397	396	4.80	5.00	5.35	5.70	5.27	6.29	6.18
21	2176	900	698	603	495	426	419	412	4.52	4.68	5.07	5.51	5.03	6.10	6.02
22	....	914	712	604	491	424	414	415	4.17	4.33	4.84	5.39	4.76	6.02	5.90
23	....	838	617	507	459	401	397	399	4.13	4.21	4.56	4.96	4.68	5.51	5.39
24	....	867	653	525	468	412	405	412	3.93	4.09	4.48	4.92	4.48	5.51	5.39
25	....	961	711	541	491	426	423	423	4.92	5.00	5.59	6.10	5.63	6.85	6.73
26	....	982	752	563	504	438	437	426	5.35	5.43	5.98	6.45	5.90	7.08	6.88
27	....	819	610	478	439	401	387	379	2.71	2.83	3.03	3.42	3.07	3.62	3.62
29	....	689	570	423	401	385	383	367	2.71	2.87	3.18	3.66	3.11	4.05	3.97
30	....	703	612	471	435	396	396	369	2.83	2.91	3.18	3.58	3.18	3.81	3.74
33	....	792	599	464	428	399	388	376	3.18	3.26	3.54	3.93	3.58	4.25	4.21
34	....	716	550	455	414	381	379	374	2.44	2.55	2.71	2.95	2.71	3.14	3.11
35	....	714	549	453	408	381	378	376	2.28	2.40	2.59	2.83	2.63	3.11	3.07
37	....	727	545	442	405	379	377	369	2.48	2.51	2.75	2.99	2.79	3.18	3.18
39	....	649	496	397	388	367	363	360	1.92	2.04	2.24	2.48	2.12	2.83	2.79
40	....	603	446	370	367	352	354	360	1.96	2.12	2.24	2.40	2.20	2.63	2.63

Tests 1–19 inclusive with mixed coals from mines No. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

TABLE 6 1000-H.P. BOILER, GAS ANALYSIS

Test No.	Analysis of Gas in First Pass. Top of 14th Tube			Analysis of Gas before Breeching. Gas Leaving Boiler		
	Per Cent CO_2	Per Cent 0	Per Cent CO	Per Cent CO_2	Per Cent 0	Per Cent CO
1	11.70	5.42	0.160	10.32	7.44	0.035
3	10.70	6.60	0.166	No record		
4	10.22	5.35	0.040	10.10	Machine record	
5	10.47	5.83	0.070	9.39	Machine record	
6	10.15	7.25	0.050	9.00	Machine record	
7	10.30	7.53	0.000	9.00	Machine record	
8	10.64	7.54	0.064	10.38	8.17	0.030
9	10.62	7.88	0.000	10.16	8.41	0.025
10	11.08	7.85	0 058	10.02	8.60	0.000
11	10.90	7.66	0 030	10.30	Machine	
12	12.13	6.37	0.040	11.20	Machine	
13	12.28	4.98	0.250	10.50	Machine	
14	12.09	6.34	0.070	10.59	7.97	0.012
15	12.21	6.05	0.020	10.44	8.00	0.002
16	11.75	6.42	0.000	10.54	8.06	0.057
17	12.02	6.29	0.100	10.68	7.73	0.046
18	11.42	7.40	0.020	10.22	8.52	0.004
19	12.19	5.94	0.067	10.77	7.86	0.022
21	11.89	6.69	0.000	10.75	8.22	0.000
22	12.59	6.13	0.071	10.89	7.87	0.059
23	11.40	7.31	0.057	10.15	8.44	0.029
24	11.91	6.61	0.022	10.00	8.47	0.000
25	13.70	4.30	0.016	11.66	6.03	0.0098
26	13.35	4.60	0.025	11.35	6.78	0.000
27	11.67	7.00	0.000	10.17	8.30	0.000
29	11.07	7.63	0.000	10.01	8.65	0.000
30	11.43	7.07	0.033	9.84	8.83	0.011
33	10.12	8.68	0.000	9.24	9.56	0.000
34	9.91	8.99	0.090	9.31	9.44	0.026
35	10.81	7.84	0.000	9.39	9.45	0.000
37	10.60	8.12	0.000	9.42	9.15	0.000
39	10.39	8.22	0.000	8.84	9.83	0.000
40	10.69	7.96	0.000	9.43	9.43	0.000

Tests 1–19 inclusive with mixed coals from mines No. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

TABLE 7 1000-H.P. BOILER. COAL ANALYSIS. PERCENTAGES BY WEIGHT CALORIFIC VALUES

Test No.	Proximate Analysis Dry Coal			Moisture in Coal as Fired	Carbon in Dry Refuse, per Cent	B.t.u. per Lb. Dry Coal	B.t.u. per Lb. Combustible.	Per Cent Sulphur in Dry Coal
	Volatile Matter	Fixed Carbon	Ash					
1	28.99	62.00	9.01	2.88	15.49	14334	15987	2.59
3	29.50	62.53	7.97	3.20	10.30	14541	16031	2.94
4	29.97	62.53	7.50	3.26	10.09	14517	15894	2.40
5	29.74	62.51	7.75	3.04	13.91	14498	15926	2.62
6	28.18	63.90	7.92	2.97	8.64	14499	15922	1.83
7	28.86	63.31	7.83	3.78	1.44	14526	15945	1.92
8	28.85	63.39	7.76	4.29	3.90	14493	15913	2.49
9	29.22	62.97	7.81	2.97	3.39	14445	15881	2.57
10	29.24	61.76	9.00	2.60	9.69	14289	15922	2.57
11	29.63	63.29	7.08	2.95	10.93	14624	15922	2.44
12	29.22	62.76	8.02	2.97	17.91	14454	15910	2.24
13	29.70	62.03	8.27	3.29	18.08	14396	15922	3.03
14	29.78	62.98	7.24	2.80	14.75	14602	15922	2.20
15	29.98	62.35	7.67	3.11	23.79	14467	15869	2.32
16	29.79	63.00	7.21	3.41	23.52	14590	15922	2.15
17	30.20	63.00	6.80	3.74	21.10	14618	15860	2.07
18	29.42	63.37	7.21	3.40	20.05	14645	15971	2.29
19	30.57	62.34	7.09	3.15	24.50	14471	15761	2.31
21	28.40	64.13	7.47	3.60	28.73	14518	15864	1.77
22	28.36	63.75	7.89	4.10	21.48	14462	15883	1.83
23	28.31	62.65	9.04	4.90	17.80	14263	15885	1.95
24	28.40	64.05	7.55	3.99	25.76	14522	15883	1.90
25	28.15	63.76	8.09	3.77	30.68	14415	15858	1.56
26	28.30	62.70	9.00	4.15	29.18	14290	15903	1.83
27	27.86	63.20	8.94	4.27	14.94	14284	15883	1.88
29	28.12	62.78	9.10	4.73	14.37	14254	15885	2.06
30	28.20	63.45	8.35	3.76	16.23	14430	15947	2.18
33	28.56	63.06	8.38	3.70	20.14	14380	15883	1.94
34	28.27	63.21	8.52	4.30	7.99	14376	15915	2.08
35	28.66	63.40	7.94	4.68	7.07	14455	15883	1.91
37	28.11	63.73	8.16	4.61	7.59	14409	15871	1.79
39	28.92	63.57	7.51	4.12	4.69	14515	15883	2.16
40	29.27	62.51	8.22	4.23	5.09	14378	15874	2.35

Tests 1–19 inclusive with mixed coals from mines No. 24, 25, 27 and 29.
Tests 21–40 inclusive with coal from No. 29 mine only.

THE V-NOTCH WEIR METHOD OF MEASUREMENT

By D. Robert Yarnall

ABSTRACT OF PAPER

This paper explains the V-notch weir method of measurement and a convenient form of apparatus, the Lea recorder, which can be used for recording graphically and continuously and with a high degree of accuracy rates of flow over V-notches and other weirs. It is shown why the V-notch is the most accurate and convenient form of weir for general testing purposes.

The paper also gives details of tests made with the apparatus measuring hot boiler feedwater at the Dartmouth College power plant during both winter and summer load conditions. The results of these tests show the great accuracy of the V-notch at low rates of flow as well as high.

Results of additional tests recently made in Philadelphia and near New York are given which together with the Dartmouth College tests show that the V-notch method of measurement gives results at high, intermediate and low rates of flow which can be relied upon to within one per cent.

THE V-NOTCH WEIR METHOD OF MEASUREMENT

By D. Robert Yarnall, Philadelphia, Pa.

Member of the Society

Attempts to measure accurately and to record automatically the flow of water through pipes or channels have been made from time to time with varying results as to accuracy and convenience. This paper will attempt to show results obtained by the simple and practical V-notch method of measurement, which though not new, seems now to be meeting with marked success, both abroad and in America, in power plant measuring problems.

2 Two tests of the accuracy of this method of measurement have recently been made in America and the data collected are here presented to show the degree of accuracy attainable.

3 The instrument employed in the tests is the Lea V-notch recorder of which a description follows:

4 This apparatus depends on the laws governing the flow of water through weirs, which laws have been found to work with extraordinary accuracy. It is only comparatively recently, however, that they have been turned to account in measuring widely varying rates of flow, both hot and cold, met with in power plants.

5 The most common form of weirs is the plain horizontal sill and the sharp edge rectangular notch. The former of these has the full width of the stream or channel and is without end contraction, the formula used in connection with it being that of Francis

$$\text{cubic feet per second} = 3.33\, LH\sqrt{H}\ .$$

where

$L =$ the width of the weir in feet.

$H =$ the head of water passing over it in feet.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

6 The second weir is a sharp edge rectangular notch of less width than the channel or stream and therefore has end contraction. The formula used in this is

$$\text{cubic feet per second} = 3.33\,LH\,\sqrt{H \times C}$$

where

L and H are the same as before.

C = a constant which depends on the ratio of the width of the weir to the width of the channel.

7 Although these two formulae give a fair degree of accuracy there is another case in which the accuracy is much greater and approaches very nearly 100 per cent. This is the sharp edge V-notch of Prof. James Thomson, the formula of which is

$$\text{cubic feet per minute} = 0.305\,H^2\,\sqrt{H}$$

where

H = the height of the notch in inches, the angle of the notch being 90 deg.

8 Table 1 gives the flow of water through 90-deg. V-notches for each inch of head, up to and including 15 in., calculated by Thomson's formula.

TABLE 1 FLOW THROUGH 90-DEG. V-NOTCH WEIRS

Depth in Notch, In.	Flow per Hour, Lb.	Depth in Notch, In.	Flow per Hour, Lb.
1	1.140	9	277,960
2	6.480	10	361,740
3	17.830	11	459,030
4	36.610	12	568,720
5	63.940	13	694,710
6	100.860	14	836,110
7	148.290	15	993,510
8	207.060		

9 The section of the flow through the V-notch is at all times the same shape, though the area may vary, and this constancy of form tends to simplify the formula and make it accurate. This form of notch is also especially adapted for measuring smaller quantities of water than the rectangular weirs, as shown by the curves in Fig. 1 which give a comparison of the accuracy of the different types of weirs at different rates of flow. Another property of the notch is that its angle may be less than 90 deg.

without impairing its efficiency, which enables it to be used to measure very small quantities of water.

10 It was for these reasons, as well as for its accuracy, that the V-notch was adopted for use in connection with the Lea recorder. A sectional view of the apparatus is shown in Fig. 2. It will be seen that a float spindle, rigidly attached to a float which rides on the surface of the water flowing over the V-notch, passes up through the bottom of the instrument case. A rack on this spindle gears into a small pinion upon the axis of a drum revolving between centers.

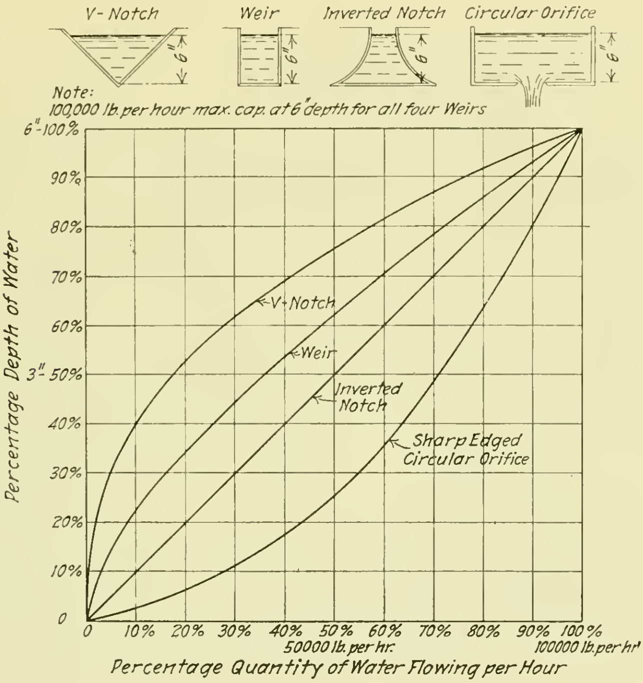


Fig. 1 Curve showing Comparative Accuracy of Different Types of Weirs under Different Rates of Flow

11 Upon the body of the drum is a screw thread, Fig. 3, the contour of which is the "curve of flow" for the V-notch in connection with which the recorder is used, and as the flow through the notch increases rapidly with the depth, the pitch of the screw increases in the same proportion. Above this drum is a horizontal slider bar, supported on small pivoted rollers and carrying an

arm, at the upper end of which is a pen or pencil point in contact with a paper chart upon a clock driven recording drum, which revolves once in 24 hours (Fig. 4). As the float rises or falls the drum spiral is rotated, and its motion is imparted to the slider bar and pen arm by means of a saddle arm projecting from the latter and engaging at its lower end with the screw thread. On the body of the spiral drum is a graduated scale adjacent to the screw thread and computed in pounds per hour. It is thus seen that the drum serves the double purpose of rectifying the motion of the pen so that it moves equal distances for equal increments in the rate of flow; and of providing a magnified scale for making an accurate observation of the rate of flow at any moment.

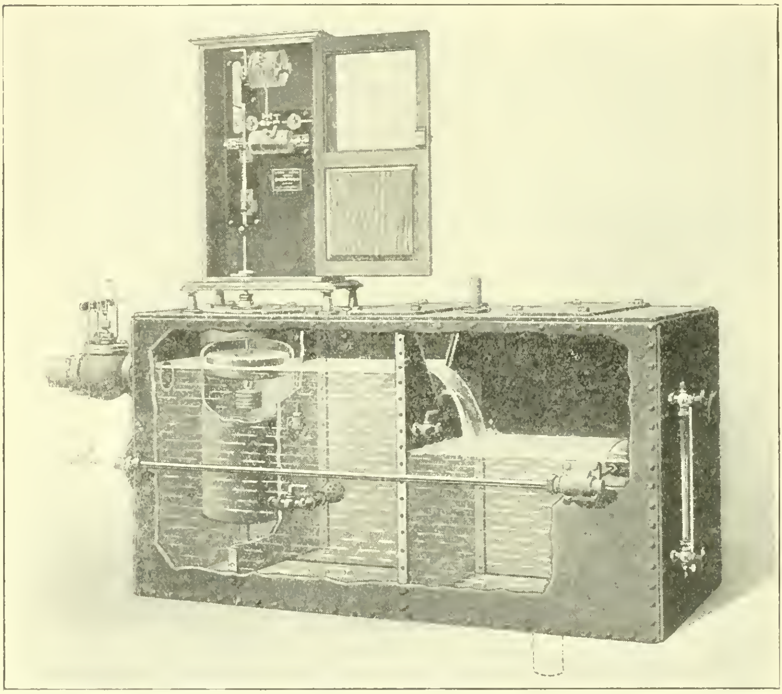


Fig. 2 Sectional View of Apparatus used in Tests

12 The apparatus has the following advantages:

a It is simple and hence dependable.

b Its operation is continuous.

c There are no moving parts in contact with the flow except the float, which is in a chamber by itself and not in the path of the flow.

d Collections of sediment and deposits of scale do not affect the accuracy.

e Owing to change of density of the liquid being measured due to changes in temperature, the float which actuates the recorder is immersed more or less, depending upon this change of density, and the recorder and the float are so constructed that they compensate automatically for these changes in temperature over a wide range.

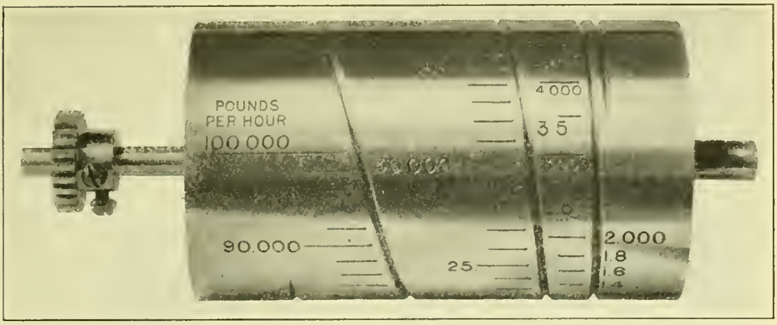


Fig. 3 Detail of Recorder Drum showing Graduated Curve of Flow

f By making the tank of ample proportions a considerable interval of time takes place between changes in the head on the weir, so that the recording apparatus has a chance to adjust itself to changes in the rate of flow, thus giving a chart very easy to measure with the planimeter.

g It is always placed on suction side of pump, hence it is available at all times for inspection, cleaning and repairs.

13 Like the Venturi meter and the Pitot tube, the V-notch recorder is a continuous flow meter, but unlike these it is not dependent for its accuracy on velocity changes, and hence its record is not subject to the fluctuations due to the sudden changes

of velocity head when these meters are in service, measuring rapidly changing rates of flow.

14 Perhaps the greatest advantage of this method of measurement is found in the way by which it may be checked independently of the recording mechanism. The operator may construct alongside of his recorder an independent "hook" gage or use the graduated vertical scale and pointer on the extended float rod in the instrument, and can observe from time to time the number of inches head on the weir. By substituting these values in Thomson's formula in Par. 7, the rates of flow as shown on the recorder chart may be checked quickly and accurately. Thus the operator need not depend upon the recorded result until he is sure that it agrees with the figured result from his hook gage readings. The author believes, therefore, that the V-notch with its tank is the most satisfactory testing set that could be devised for measuring large and small rates of flow, hot and cold, as well as fluctuating or uniform conditions of flow.

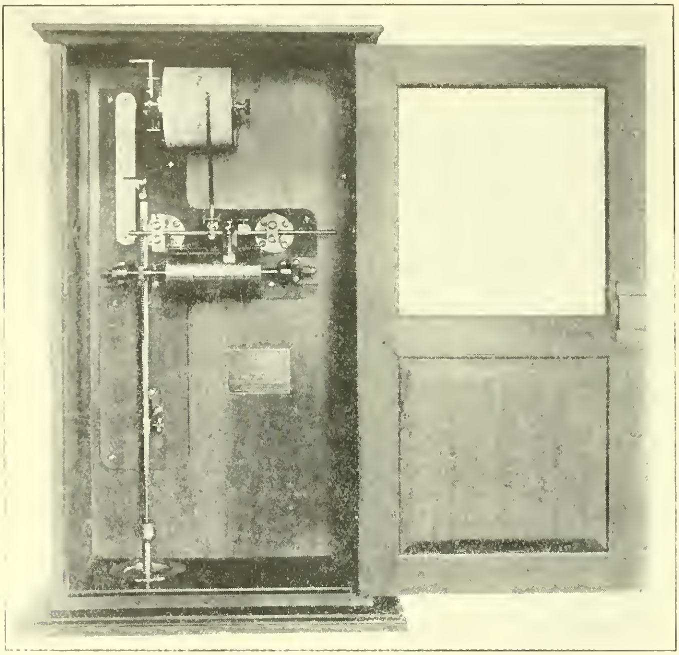


Fig. 4 Detail of Recorder

15 For results the following reports of tests are given, and as clearly as possible the actual test conditions which prevailed explained.

16 With a view to checking the accuracy of this apparatus and the V-notch for the measurement of boiler feedwater a series of tests were made at the Dartmouth College power plant, at both winter load and summer load. These tests were conducted by the engineers for the college, Messrs. R. D. Kimball Company, through their representative J. H. Browne, assisted by the writer.

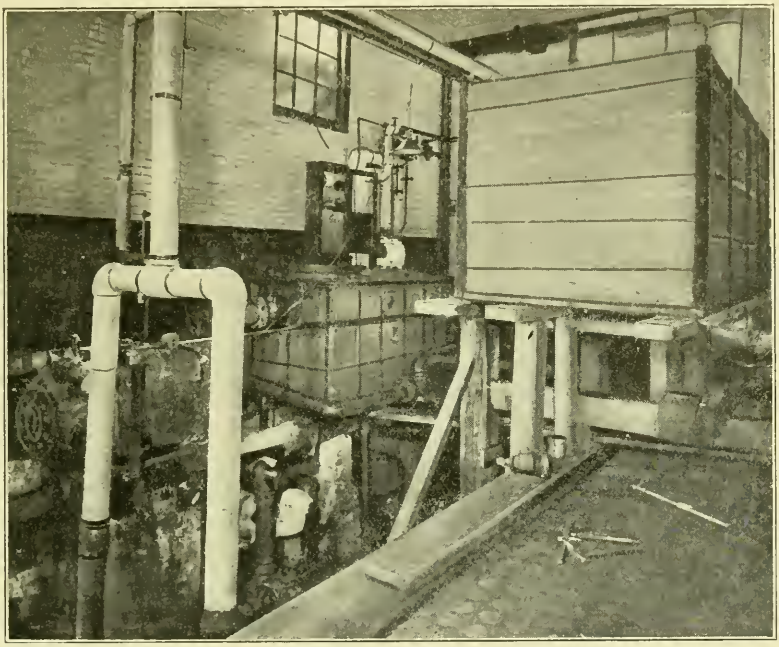


FIG. 5 ARRANGEMENT OF TESTING APPARATUS FOR THE DARTMOUTH COLLEGE TEST

DARTMOUTH COLLEGE WINTER TEST

17 Fig. 5 is from a photograph of the testing apparatus and meter. On the right-hand side is a large wooden tank divided into two parts by a vertical partition. The size of this tank was $54\frac{5}{8}$ in. deep by $54\frac{1}{4}$ in. wide by one section $59^1/_{16}$ in. long and the other section $59\frac{1}{8}$ in. long, inside measurements. To the

left is the cast-iron V-notch tank and mounted above is the recording apparatus. The inlet of the meter is controlled by a 6-in. balanced float valve operated by a float riding on the surface of the storage section of the notch tank below the weir, so as to prevent flooding in the event of the feed pump slowing down or stopping (Fig. 2).

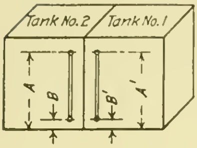
18 The return water from all of the buildings of Dartmouth College is collected in the receiver shown at the left of the meter tank. The make-up water also enters this receiver, which is under about 5 lb. back pressure. The 4 in. line discharges from this receiver into either one of the sections of the wooden measuring tanks, the water flowing by gravity from this tank through the meter to the feed pump and is thence pumped into the boilers in the adjoining boiler plant.

19 The two sections of the wooden tank were provided with independent vertical glass columns ¾ in. in diameter. Just back of and touching each glass column was provided a scale graduated in 16ths of an inch. The two tanks were carefully calibrated by actually weighing the value of 1 in. in height of the water in the tank at a given temperature, this value being independently computed as noted in the tables. It was found during the test that the meniscus of the column at the upper and lower reading could be read with great accuracy because of the careful construction of the column and the graduation of the scale.

20 It is to be noted that this test was made at the normal running load and took care of all of the natural fluctuations in the returns from the heating system. The test was conducted when the outside temperature was only slightly above zero most of the time, hence the heating system was being taxed to its utmost.

21 The test was run under exact working conditions of the plant at heavy load, which fluctuated badly (Fig. 6A). These fluctuations in the load were caused by poor control of the return water from the new gymnasium which was at a lower elevation than the power plant and the accumulated returns from the heating system were intermittently pumped up into the power plant by a float controlled motor driven pump. Only very small water storage was provided in the receiver, hence the fluctuating boiler feed conditions shown: under these conditions the results of the test are remarkably accurate.

TABLE 2 TEST OF V-NOTCH RECORDER AT DARTMOUTH COLLEGE, MARCH 23, 1912



$A - B = C$

$A' - B' = C'$ Correct Quantity $= \Sigma (C + C')$

METER CAPACITY, 50,000 LB. PER HR.; 6-IN. RECORDER; 90° V-NOTCH 6 IN. DEEP

	Tank No. 1			Tank No. 2			
Time	*A*	*B*	*A–B*	*A'*	*B'*	*A'–B'*	Temperature, Deg. Fahr.
9.00	$37\frac{3}{16}$	7⅝	$29\frac{9}{16}$	42⅝	5	37⅝	164
9.22	42½	3⅝	38⅞	34½	3	31½	
9.41	32⅞	3⅛	29¾	39⅞	2¾	37⅛	180
9.50	41⅛	3¼	37⅞	42¾	8¾	34	
10.15	43⅞	7⅛	36¾	43⅛	3⅜	39¾	180
10.40	42	2¾	39¼	39⅞	4½	35⅜	
11.06	40¼	2	38¼	42¾	3½	39¼	170
11.40	44	3⅞	40⅛	44	7¼	36¾	
12.00	43½	3	40½	43¼	3	40¼	160
12.27	44	5⅜	38⅝	44⅝	2½	42⅛	
1.00	44	2	42	44	1¾	43¼	195
1.25	44⅜	2¼	42⅛	44¼	2⅛	42⅛	
1.50	46¼	4⅜	41⅞	44½	2⅞	41⅝	195
2.15	43	2¼	40¾	44	⅝	43⅜	
2.52	39⅜	3⅜	38	43⅞	2¾	41⅛	194
3.25	44	1⅛	43⅞	41⅛	3½	37⅝	190
4.00			Total....$615\frac{3}{16}$ in. or 615.18 in.			Total.. 621⅞ in. or 621.87 in.	

Tank calibration by actual weight at 175 deg. fahr.:

In tank No. 2, 1 in. = 117.0 lb.
In tank No. 1, 1 in. = 115.2 lb.
615.16 × 117.0 = 71976
621.87 × 115.2 = 71640
Total...... 143616
Error due to tank leakage 225 143841

From planimeter measurement of chart (see Fig. 6A)

Quantity, lb. = Area × Constant
4.04 × 36312 = 146700.48

Correction for zero adjustment of recorder............. 1,162 145538

Recorder error (high) = 1697
Recorder error, per cent = 1.18

DARTMOUTH COLLEGE SUMMER TEST

22 The second test was made in the summer to check accurately the claim that the meter is as accurate at low rates of flow as at high. Table 3 substantiates this claim as the accuracy was even greater (0.55 per cent) than that at the winter load. This due to the more even rate of flow (see low reading) which in turn is due to the gymnasium heating system not being in service.

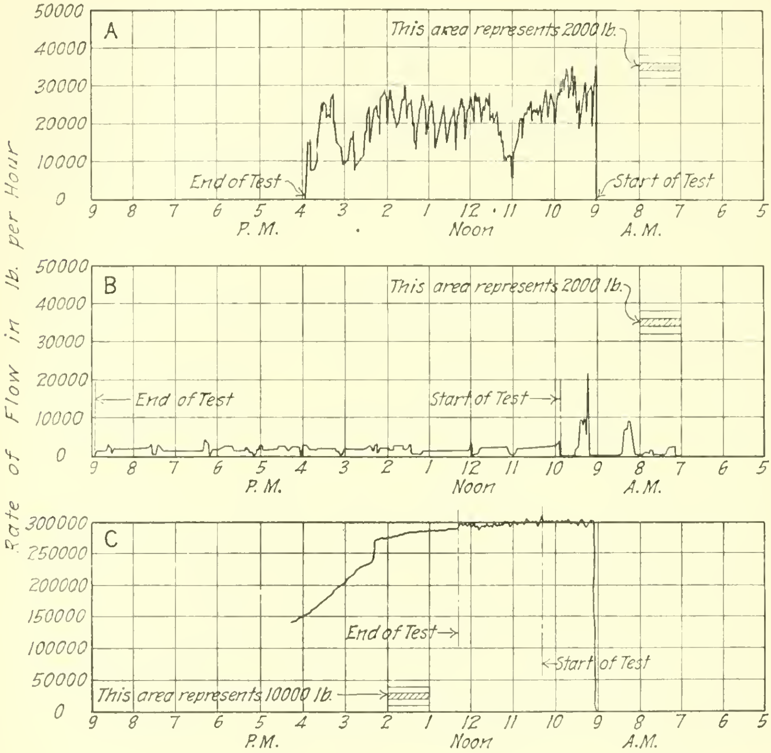


Fig. 6 Recording Charts

TESTS ON V-NOTCH METER IN PHILADELPHIA, AUGUST 9, 1912

23 Figs. 7 and 8 show the arrangements for another test conducted recently in exactly the same way as the tests at Dartmouth

College except that the water was measured on the discharge side of weir instead of the intake side. In this test, however, the water being measured was at 73 deg. while the Dartmouth tests were run at an average of 190 deg. fahr.

24 The V-notch meter was mounted for convenience on a platform car on the coal siding shown in Fig. 8. The car during the test was rigidly jacked to prevent vibration.

TABLE 3 TEST OF V-NOTCH RECORDER AT DARTMOUTH COLLEGE, AUGUST 2, 1912

Tank No. 1				Tank No. 2			
Time	Full, In.	Empty, In.	Weight	Time	Full, In.	Empty, In.	Weight
10.52	32.0	....	......	9.43	44.81	6.50	4495.3
11.47	....	1.68	3502.3	11.41	32.44	7.81	2890.0
1.03	40.0	0.875	4520.8	2.03	33.81	1.75	3761.9
2.55	34.5	2.00	3655.4	3.54	41.50	1.63	4680.1
4.55	41.25	9.125	3712.0	6.00	45.31	1.19	5177.0
7.15	40.25	7.06	3880.8	8.00	41.38	29.87	1349.4
		Total.......	19271.3				22353.7
							19271.3
					Total............		41625.0

Chart area = 0.570
Constant = 36312. lb. per sq. in.
Product = 20697.84
As 90 V-notch is used multiply by 2 as drum is cut for ½ 90........ 41395.7
Error............. 2.293
Result............. 0.55%

Below the notch tank are shown two large cast-iron tanks, made of sectional plates 10 ft. 8 in. long by 8 ft. wide by 6 ft. 2 in. high, inside dimensions. The water from the meter could be discharged into either one of these measuring tanks by throwing over a tipple without interrupting the flow. These measuring tanks were provided with carefully constructed glass water columns with accurately graduated scales back of the glass. Baffle plates were constructed within the measuring tanks to prevent wave motion and resulting fluctuations in the water columns. Below the measuring tanks is shown the large storage supply tank 22 ft. long by 8 ft. wide by 6 ft. 2 in.

high, also built of cast-iron sectional plates, into which the upper tanks were in turn drained. The measuring tanks were very accurately calibrated by weighing on a corrected scale a unit charge of 100 lb., marking the scale through a certain range near the top and bottom of the tank for each unit charge of 100 lb.

25 The test as will be observed was run at approximately the rate of 300,000 lb. per hr. This was the maximum capacity of the recorder under test. A chart record was obtained during the test which was measured carefully in order to get the quantity for comparison with the quantity measured in the tanks below the weir tank. The recorder had a drum with a curve of flow for use at a temperature of 212 deg. As the water during the test was 73 deg. it was necessary to make a correction for this large difference in temperature. This correction has been applied in the data of the test as will be noted. Correction is also made for inaccuracy of scale as found by calibration.

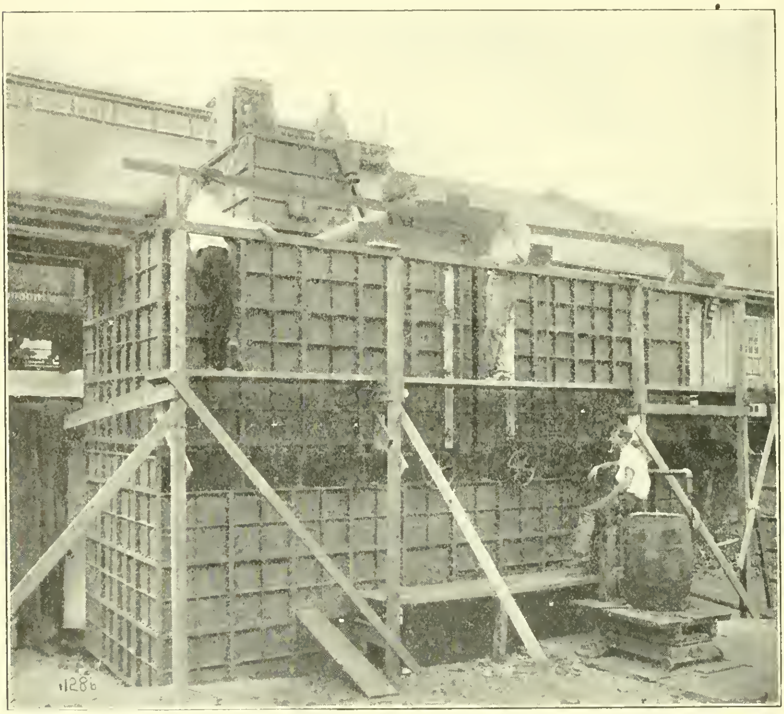


Fig. 7 Arrangement of Testing Apparatus for Philadelphia Test

Duration of test = 1:57.86 = 1.9643 hour.
By scaling chart = 1.09375 in. = 1.9643 hour.
Hence recorder clock driving chart is correct.

Then 1 in. of chart = $\frac{1.9643}{1.09375}$ = 1.796 hours.

By scaling chart (vertically) 2.5 in. = 300,000 lb. per hr.
Therefore 1 in. = 120,000 lb. per hr.
Therefore 1.796 × 120,000 = 215520 lb. = 1 sq. in. in Fig. 6C.

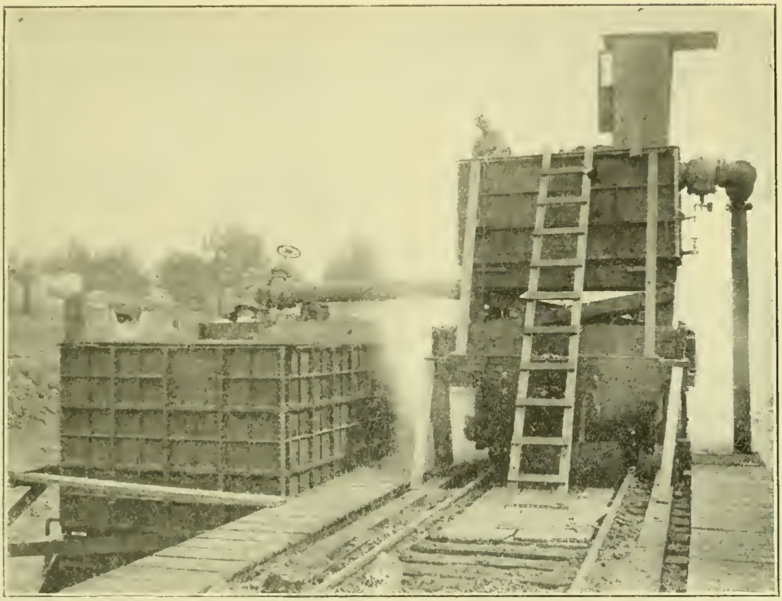


FIG. 8 ARRANGEMENT OF TESTING APPARATUS FOR PHILADELPHIA TEST

26 Integrating chart area under curve = 2.70 sq. in.
Hence 215520 × 2.7 = 581,904.0 lb. = recorder record.

27 Error in scales: each 400 lb. water as graduated at gage glass on tank is actually 400.75 lb. by correct weight, each pound of water passing through the meter cold (about 68 deg.) will increase in volume to such an extent when hot as to require the addition of 3½ per cent of the indicated chart record to the value given by the chart when the cam is cut for a range of from 175 deg. to 225 deg., average 200 deg.

Correction for scales = 1.001875
Correction for temperature = 1.035

581,904 × 1.035 =	602270.64	chart weight as corrected
595,750 × 1.001875 =	596867.03	corrected weight by tanks
Difference =	5403.61	recorder lead
Result =	0.9 per cent	recorder lead

TEST ON V-NOTCH METER NEAR NEW YORK, APRIL 1912

28 It is to be regretted that the series of tests made in April 1912 on a 400,000 lb. per hour V-notch meter cannot be described in full in this paper; the results in brief form are as follows:

29 This series of tests differs from the two tests just described in that the water was actually weighed on large platform scales, and hence no correction for temperature was necessary.

TABLE 4 TEST MADE NEAR PHILADELPHIA, PA., AUGUST 9, 1912

Consecutive Time	Elapsed Time	No. of Tank	North Bottom Reading	North Top Reading	Net North	South Bottom Reading	South Top Reading	Net South	Temperature
10:25.10*									
10:30.10	:05.00	1				26200	410	25790	
:35.07	:04.97	2	26250	100	26150				72
:40.09	:05.02	3				25875	150	25725	
:45.54	:05.45	4	26150	210	25940				72
:50.54	:05.00	5				25950	170	25780	
:55.64	:05.10	6	26150	220	25930				72
11:00.74	:05.10	7				26160	230	25930	
:05.87	:05.13	8	26330	220	26110				72
:10.96	:05.09	9				26220	210	26010	
:16.06	:05.10	10	26025	220	25805				72
:21.17	:05.11	11				25950	220	25730	
:26.29*	:05.12	12	26025	200	25825				73
:31.41	:05.12	13				26100	180	25920	
:36.53	:05.12	14	26075	200	25874				73
:41.66	:05.13	15				26030	200	25830	
:46.87	:05.21	16	26300	200	26100				
:52.00	:05.13	17				26025	160	25865	73
:57.20	:05.20	18	26000	180	25910				
12:02.39	:05.19	19				26450	160	26270	73
:07.53	:05.14	20	26000	210	25790				
:12.68	:05.15	21				26030	190	25840	73
:17.88	:05.20	22	26025	180	25845				
:22.96*	:05.08	23				26000	220	25780	74
1:57.86	:117.86		287420	2140	285280 310470	312990	2520	210470	74
			Total weight.........		595750				

* Marked chart and started to fill south tauk.

5-Hour Test on V-Notch Meter, April 19

Water by actual weight, lb.	291965
Water by recorder, lb.	290525
Lead of scales, lb.	1440
Lead, per cent	0.49

8-Hour Test on V-Notch Meter, April 20

Water by recorder, lb	267260
Water by actual weight, lb	266009
Lead by recorder, lb	1251
Lead, per cent	0.46

13-Hour Test on V-Notch Meter, April 22

Water by actual weight, lb	557974
Water by recorder, lb	557775
Lead of scales, lb	199
Lead, per cent	0.03

8-Hour Test on V-Notch Meter, April 23

Water by actual weight, lb	352725
Water by recorder, lb	351501
Lead of scales, lb	1224
Lead, per cent	0.34

8-Hour Test on V-Notch Meter, April 24

Water by actual weight, lb	352725
Water by recorder, lb	352231
Lead of scales, lb	494
Lead, per cent	0.14

CONCLUSION

30 In conclusion it will therefore be noted that the V-notch method of measurement may be consistently depended upon within 1 per cent, even in view of fluctuating conditions of the rate of flow and also fluctuating temperatures. By using ¼ 90 degree notch weir, still greater accuracy may be obtained, and for all particularly accurate tests either ½ 90 degree or ¼ 90 degree weir plates are recommended.

31 Attention is here called to an able discussion of the V-notch with results of tests made by James Barr, B.Sc. Carnegie Research scholar at the James Watt Engineering Laboratories, Glasgow University, 1907–1909. The synopsis of his results appear in London Engineering, April 8 and 15, 1910 editions.

DIMENSIONS OF BOILER CHIMNEYS FOR CRUDE OIL

AT SEA LEVEL AND AT VARIOUS ALTITUDES, WITH CORRECTING FACTORS FOR FUEL ECONOMIZERS

By C. R. Weymouth

ABSTRACT OF PAPER

Chimney tables applicable to boiler plants using coal fuel give dimensions larger than necessary for oil fuel. It is impossible to compile a single table of correct chimney capacities applicable to different types of boilers, operating conditions, etc., for oil fuel.

This paper sets forth the elementary principles for the determination of the dimensions of chimneys, with special reference to the use of crude oil; it also presents data regarding the draft necessary for burning oil under Babcock and Wilcox boilers of various sizes, at various ratings, and a table of chimney capacities applicable to certain sizes of Babcock and Wilcox boilers.

The operating fuel economy of a crude oil plant is largely influenced by the excess air supply due to excessive draft, and the author recommends the use of lower chimneys than customary, particularly for uniform load plants, not subject to overload, as affording a means of chimney regulation of fuel economy.

Errors are shown in previously accepted formulae for variation of chimney proportions for operation at different altitudes, and correct formulae are given.

Data are given regarding the frictional resistance of fuel economizers, and a table presented showing ratio of reduction of chimney capacity, due to cooling of gases within economizer, and the resistance of economizer. Dimensions of steel chimneys are given for minimum first cost, and selection of type of chimney suitable for quick starting reserve plants is discussed.

DIMENSIONS OF BOILER CHIMNEYS FOR CRUDE OIL

AT SEA LEVEL AND AT VARIOUS ALTITUDES, WITH CORRECTING FACTORS FOR FUEL ECONOMIZERS

By C. R. Weymouth, San Francisco, Cal.

Member of the Society

The necessary heights for chimneys for fuel oil plants are much less than ordinarily supposed when the boilers are operating at rating and considerably greater than usually supposed when there are heavy overloads. A chimney of undue height causes an excessive quantity of air for combustion and permits an excessive load on the boilers, both resulting in a large waste of fuel.

2 When the chimney height is limited to that necessary for economical air supply at the desired rate of load, it will be impossible for the most careless fireman to cause a material loss of economy, either by overload or excess air supply, or abuse of boilers due to overload. In this sense the chimney may become an important and inexpensive means for boiler plant regulation and in many ways a safeguard against careless firing. Such usage is, of course, of greatest effect in plants operating at uniform load.

3 Although the writer had previously intended investigating this subject, it was not until the San Francisco earthquake of April 18, 1906, had considerably reduced the height of most masonry chimneys that he was led to an active collection of chimney data. Many of the results apparently were contradictory. Certain chimneys reduced to a height of 30 ft. gave the usual boiler capacity and others, reduced to a height of 75 ft., showed under certain conditions of service a decrease in boiler capacity.

4 If the total or potential draft of a chimney were available

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The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

for the flow of gases through the chimney and equal to the frictional resistance within the chimney, the capacity of the chimney would then be a maximum. On the other hand, if a large percentage of the draft of a chimney were maintained at the base it is self evident that but little draft would be available to induce a flow of gases; and the greater the draft power at the base of chimney, the less the capacity of chimney in weight of gases discharged per unit of time.

5 In this respect a chimney of given diameter and height may be likened to a centrifugal fan of given diameter operating at a fixed rate of revolutions. When operating on free air discharge the capacity of the fan is very large, but when operating against an increasing pressure the rate of discharge rapidly decreases. No one would think of computing a table showing fan capacities without specifying the size of fan and the working pressure.

6 In the same sense a table of chimney capacities is incomplete which does not designate the working draft at the base of chimney; particularly for oil fuel, because of the rapid increase in draft resistance as boilers are overloaded. For an assumed set of conditions, a chimney, as with a fan, has a definite capacity, but no overload capacity.

7 Chimneys should therefore be selected, not with respect to the rated horsepower of the connected boilers, but after consideration of the maximum boiler overload and the draft resistance of boilers at overload.

8 While most chimney tables are reduced to a boiler horsepower basis, for convenience, the correct designation of capacity is the weight of gases discharged per unit of time. It follows that in determining the weight of gases from the boiler horsepower, two important factors enter: the ratio of air for combustion to that chemically required, and the efficiency of boilers at the maximum load.

9 Knowing the weight of gases to be handled, it is comparatively an easy matter to determine from existing formulæ the frictional loss due to a given chimney and flue. There are, however, but few published data indicating the draft resistance of various types of oil fired boilers under various rates of firing.

10 Practical experience has shown that altitude is an important factor affecting the capacity of boiler chimneys. Certain rules of thumb are now employed, and certain formulæ have been

incorrectly derived. The writer's treatment of this question is also given in this paper.

WEIGHT OF CHIMNEY GASES

11 In a given plant, the total weight of gases per hour to be handled by chimney is dependent upon the rated capacity of the boilers, their maximum momentary overload capacity, the necessary percentage of excess air for a given type of boiler and furnace, and the boiler efficiency for the conditions assumed.

12 Tests have indicated the efficiencies of boilers, the temperatures of waste gases, and the minimum air required for combustion. In every-day practice the negligence and ignorance of firemen, the disarrangement of furnace brickwork, the carbonization of burners and the improper adjustment of dampers make it necessary to provide for an excess of air over the minimum air supply obtainable in tests, depending on the type of boiler, the character of service, skill of firemen, etc.

13 For the best types of water-tube boilers fitted with Peabody furnace and operating at rating or at overloads, the writer has assumed as ample margin for good practice 50 per cent excess air over chemical requirements at chimney inlet. In determining draft resistance of boilers, 50 per cent excess air through the setting has been assumed for boilers at rating, and slightly less at 50 per cent overload, as explained below.

14 For poorly designed and operated furnaces, 50 per cent excess air is an insufficient allowance and the chimney proportions given in the tables must be modified by the designer to meet actual conditions.

15 When new, a boiler plant will require a minimum of air for combustion and usually there will be but little greater weight of gases passing up the chimney than into the furnaces. Allowance must be made for leaky boiler settings, long masonry flues, etc., but with steel flues air infiltration should be negligible. The writer recalls a test of an old plant having long brick flues, where the rate of flow of gases up the chimney was about double that into the connected boiler furnaces; and as a result of this leakage of cold air, the chimney temperature was very much lower than that of the gases leaving the boilers.

16 With expert firing it is possible to reduce the air supply to nearly the theoretical minimum, both at rated load and at overloads. At rating, the relatively large furnace volume makes it

possible for considerable amounts of air to pass through the furnace without entering the zone of combustion: but as the load is increased the flame more completely fills the furnace and, except with a deliberate wastage, a larger portion of the air admitted commingles with the flames and aids combustion.

17 In the computations which follow the writer assumes 14 lb. of air chemically required per lb. of oil and, based on 50 per cent excess, there would be admitted a total of 21 lb. of air per lb. of oil. To this the products of combustion and steam for atomization add slightly in excess of one pound, so that hereafter 22 lb. of chimney gases per pound of oil will be assumed.

18 To assist in the computation of the weight of chimney gases Table 1 is presented, giving calculated boiler efficiencies based on assumed conditions, as explained in the foot notes of the table.

19 The temperature of the waste gases and the gas analysis are more easily determined than the boiler efficiency and, based on these factors, the table assists in approximating the actual boiler efficiency and correspondingly the rate of oil consumption. Due allowance should be made for reduction of boiler efficiency due to any accumulation of boiler scale, sooty tubes, etc., after a period of use.

DRAFT RESISTANCE OF BOILERS

20 The draft resistance of a boiler depends on type, size, arrangement of heating surface, type and arrangement of superheater, if any, extent of baffling of gases, type of oil furnace, arrangement and area of air admission to boiler and to furnace, the excess air for combustion over chemical requirements and the rate at which the boiler is operated. The draft resistances of water tube boilers are ordinarily recorded in the rear of boiler, inside the setting, and close to the outlet damper. These drafts, therefore, do not include the resistance of damper and flue.

21 If a boiler be fired with wide open ash doors, the regulation of air being from the damper at boiler outlet, the draft resistance of the boiler will be a minimum. In boilers having a series of up and down passes, the draft within the boiler setting at the top of the first pass may entirely disappear and a positive pressure may result. This condition is due to the injector action of the burner pumping air into the furnace, and to the less

weight of the heated column of gases inside the boiler as compared with external column of air of equivalent height.

TABLE I MAXIMUM THEORETICAL EFFICIENCY FOR A GIVEN PER CENT EXCESS AIR SUPPLY AND FLUE GAS TEMPERATURE

Based on Stated Assumptions[1]

Assumed Per Cent Excess Air Supply	Assumed Temperature of Flue Gases, Deg. Fahr.										Assumed Per Cent Excess Air Supply
	375	400	425	450	475	500	550	600	700	800	
	Calculated Boiler Efficiency, Per Cent										
0	81.35	80.78	80.20	79.61	79.03	78.44	77.28	76.11	73.77	71.43	0
10	80.62	80.04	79.41	78.78	78.15	77.52	76.25	74.99	72.47	69.94	10
50	77.90	77.08	76.25	75.44	74.62	73.80	72.16	70.52	67.24	63.97	50
100	74.42	73.37	72.32	71.26	70.21	69.16	67.04	64.95	60.72	56.51	100
150	....	....	....	....	65.79	64.51	61.93	59.35	54.20	49.04	150
200	....	....	....	....	....	59.86	56.81	53.77	47.67	41.58	200
250	....	....	....	....	....	....	51.69	48.18	41.15	34.11	250
300	....	....	....	....	....	....	46.58	42.60	34.63	26.65	300

APPROXIMATE RELATION BETWEEN PER CENT CO_2 AND EXCESS AIR SUPPLY AS PER ASSUMPTIONS BELOW

Per cent Co_2	4	5	6	7	8	9	10	11	12	13	14	15	15.9
Per cent Excess Air.	283	205	155	120	93.0	72.2	55.5	41.9	30.6	21.0	12.7	5.6	0

[1] Values are conditioned on the following assumptions:

Average temperature of air for combustion entering the boiler 80 deg.; humidity 80%; air per lb. of oil chemically required for complete combustion, 14 lb. b.t.u. per lb. of oil as fired, 18,500; chemical composition of the oil as follows: carbon, 86%; hydrogen, 11%; sulphur, 0.8%; nitrogen, 0.2%; oxygen, 1.0%; water, 1.0%.

Per cent of excess air stated is measured at boiler outlet and consequently includes leakage through boiler setting.

The loss by radiation has been taken as 3%. This loss varies with the size of boiler, insulation, etc., being less than indicated for larger sizes of boilers and greater for smaller sizes.

An allowance of 2% for undetermined losses has been made. This quantity is subject to considerable variation and may, in exceptionally favorable instances, be as low as 0.5%.

The following constants are taken from Marks and Davis' Steam Tables: Absolute temperature $=459.6$ deg. $+ t$ deg. Heat of vaporization at atmospheric pressure $=970.4$ b.t.u. Specific heat of superheated steam at atmospheric pressure, for the range of temperature from 212 deg. to 700 deg. fahr. $=0.47$.

This table shows conservative efficiencies primarily for use in stack calculations.

22 In boilers with brick settings there may be an escape of hot gases from the top of boiler at times, giving rise to a possible loss of economy and also to annoyance of firemen from

escaping gases. Under certain conditions it is therefore necessary to close the air admission doors partially to insure a draft at all points within the boiler setting, tending to increase the total draft resistance otherwise necessary. This effect is more noticeable at loads approximating rating, or slightly below, than at overloads, where the friction of flow of air for combustion into the boiler and furnace tends to prevent greater pressure within the setting than without.

23 The analysis of over 75 tests under various conditions, on several sizes of boilers of the same general type, fails to reveal a very close relationship between the total draft resistance and the rate of load on boiler. Other conditions being equal, it is clearly evident that the draft resistance increases slightly with an increase in size of boiler for all rates of firing. A 14-tube high Babcock and Wilcox boiler has a draft resistance greater than a boiler 9 tubes high. There is a variation of the injector action of the burner with different types, arrangement of burners, arrangement of checkerwork forming the floor of furnace, position of the burner relative to checkerwork. and the form of flame relative to checkerwork.

24 Engineers and operators vary the size and arrangement of air spaces for admission to the furnace, and the length and area of air ducts leading from the air doors to the furnace.

25 The area of air admission doors has a general relationship to boiler horsepower, but changes in area if these doors are irregular between certain sizes of boilers. In a Babcock and Wilcox boiler of 600 h.p. with a certain furnace arrangement, draft resistances at rating have been noted as low as 0.08 in. and as high as 0.2 in., this variation being due mainly to increasing quantities of air supplied for combustion. A change in furnace arrangement for economical firing gave for this same boiler a draft averaging slightly over 0.10 in. in one case and less than 0.05 in. in another case, at about the same rating and efficiency.

26 The writer's data are compiled from a series of tests made especially for the determination of boiler efficiencies and not with a view to determination of stack capacities. A series of curves was plotted and an attempt made to correct for any inequalities due to the position of air admission doors, the ratio of excess air over chemical requirements, etc. Even with these corrections, for reasons outlined above. the plotted points did not fall on smooth curves.

27 Fig. 1 shows curves of average draft resistances on Babcock and Wilcox boilers of several sizes, at various rates of firing, given for minimum observed air supply. The apparent inconsistency is due to the tests having been made by different persons

28 Fig. 2 shows curves of corrected drafts to conform as nearly as may be to a rate of air supply 50 per cent in excess of chemical requirements, the curves being rationalized with respect to each other for similar furnace settings, differences being due to variation in size of boiler, etc.

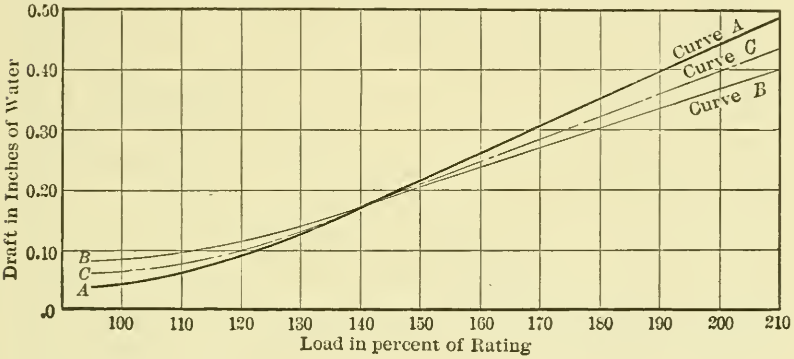

Fig. 1 Draft Resistance of Babcock and Wilcox Boilers with Peabody Furnaces burning California Crude Oil. Curves plotted through Points of Minimum Draft Corresponding to Minimum Air Supply, Test Conditions, Draft measured in Third Pass.

Curve *A*—21 wide, 14 high, 18 ft. long, 604 h.p.[1]
Curve *B*—21 wide, 12 high, 18 ft. long, 524 h.p.
Curve *C*—14 wide, 9 high, 18 ft. long, 260 h.p.

29 All drafts in Figs. 1 and 2 are measured inside of the boiler between the end of the third pass and boiler outlet.

30 It should be noted that the draft at the rear of boiler will be somewhat dependent on the height above the floor line at the point of measurement. This height will vary from about 6 to 20 ft., depending on the type of boiler, and introduces a considerable correction for the chimney height otherwise determined. It is evident that the calculated height of chimney must be measured above the point of observed or calculated draft for the boiler in question, and not from the boiler room floor line.

31 The draft resistance between the third pass and the flue will depend upon the size, type and arrangement of boiler, area

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[1] Equipped with double loop superheater.

and position of damper opening, arrangement of circulating tubes, etc.

32 The calculated draft resistance through the damper box, on the basis of velocity head, allowing ordinary coefficients for friction. does not give a result as great as the average results obtained by experiment. Draft readings are subject to error, owing to the position of draft tube as affected by velocity of gases. etc. In one instance, with a Babcock and Wilcox boiler operating at 100 per cent overload, the draft observations were the same on both sides of the boiler damper, a result manifestly in error. In a 200-h.p. Babcock and Wilcox boiler working at 67 per cent overload. with economical air supply, the draft loss between the third pass and the flue side of damper with damper wide open was 0.04 in.

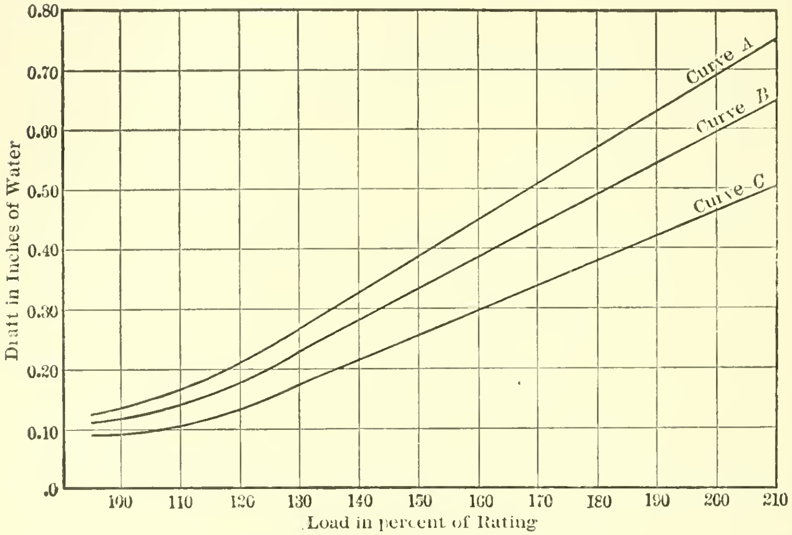

FIG. 2 DRAFT RESISTANCE OF BABCOCK AND WILCOX BOILERS WITH PEABODY FURNACES BURNING CALIFORNIA CRUDE OIL WITH DRAFT MEASURED IN THIRD PASS AND AIR SUPPLY 50 PER CENT IN EXCESS OF CHEMICAL REQUIREMENTS.

Curve *A*—21 wide, 14 high, 18 ft. long, 604 h.p.[1]
Curve *B*—21 wide, 12 high, 18 ft. long, 524 h.p.
Curve *C*—14 wide, 9 high, 18 ft. long, 260 h.p.

33 A 520-h.p. Babcock and Wilcox boiler, under good firing conditions, at slightly over rating, gave 0.03 in. loss between flue and third pass, the damper being wide open. Another test on

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[1] Equipped with double loop superheater.

boilers on the same size gave 0.02 in. draft loss, at a slight overload, economical air supply.

34 A 680-h.p. Babcock and Wilcox boiler, having overhead flue, outlet between drums, gave the same draft in flue on boiler side of damper as in the third pass, a result no doubt affected by draft differences, due to difference of height of points of measurement.

35 A series of tests on a 600-h.p. Babcock and Wilcox boiler at the Redondo plant, 14 tubes high, with a double loop superheater, corrected for 50 per cent excess air supply, gave an average curve from which the following values were taken:

Percentage of Boiler Rating	Draft Resistance, Damper only, in. Water
100	0.08
125	0.11
150	0.14
175	0.16

FRICTIONAL RESISTANCE OF STEEL FLUES, ETC.

36 It is not the purpose of this paper to present new data for frictional resistances, etc., but rather to assume the correctness of established formulæ.

37 The resistance of flues will depend upon the construction, whether of steel, brick, cement, etc., and will also vary with the character of sooty coating, etc. For circular unlined steel flues after a period of use, the writer has assumed the formula

$$D = \frac{3\ W^2\ T\ L}{d^5} \qquad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

in which

D = draft loss, in. of water.
W = weight of gases flowing per second, lb.
T = absolute temperature of gases, deg. fahr.
L = length of flue, ft.
d = diameter of flue, in.

38 The coefficient 3 is approximate only and is based on information furnished by a prominent fan manufacturer, slightly increased to allow for sooty surfaces. The same source indicates that the frictional loss of brick or cement flues would be nearly double that for clean sheet iron, but the writer believes that the loss will not increase in this ratio, owing to the fact that both surfaces will eventually have a coating of soot.

39 Compared with circular flues, the resistance of those of rectangular section is greater at the same velocity by reason of the increased wetted area, being approximately 12 per cent greater for flues of square section and 15 per cent greater for flues of 1 by 1½ rectangular section than for flues of circular section.

40 The resistance of a right angle turn is taken by most authorities as equivalent to the velocity head. Reduced to the basis of the above nomenclature, we have

$$D = \frac{2.6\ W^2\ T}{d^4} \qquad [2]$$

in which

D = draft loss due to velocity head or right angle turn

41 The formulæ for flue resistance are modified for chimneys as indicated below, the difference in formulæ being, on first thought, not entirely consistent.

42 For resistance of bends of angles greater or less than 90 deg., for diverging or contracting shapes, the designer will be compelled to modify the formulæ on the basis of standard formulæ, the factor of judgment entering largely and the final results being approximate only.

43 The resistance due to a right angle turn in the breeching or at chimney has an important effect on the height of chimney, and to minimize the loss in straight runs as well as in turns, conventional practice requires an area of breeching about 20 per cent in excess of that of chimney.

DRAFT POWER OF CHIMNEY, FRICTION LOSS, ETC.

44 The exact predetermination of the potential draft of a chimney is most difficult, as it is impossible to disassociate the frictional loss within a chimney from the draft loss due to cooling, air leakage, etc.

45 It has been pointed out in Professor Gale's paper[1] that there is a decrease in the potential draft of a chimney, due to friction and cooling of gases within the chimney, to infiltration of cold air, etc., and to atmospheric currents of air in the region of the top of chimney. It has also been shown in Mr. Babcock's discussion of the paper[2] that with a quiet atmosphere there is an

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[1] H. B. Gale, Theory and Design of Chimneys, Trans. Am. Soc. M. E., vol. 11, p. 451.

[2] Trans. Am. Soc. M. E., vol. 11, p. 482.

increment of draft due to the column of heated gases directly above the chimney; and further, that the form of the top of the chimney has a direct bearing on the chimney draft whenever the wind velocity is appreciable.

46 The writer would add that the deceleration of gases within the chimney, due to any cooling in the passage up the chimney, will result in a slight decrease of chimney friction on account of lessening of the volume and velocity of gases; and the conversion of the velocity head into a pressure head will also tend to overcome the downward atmospheric pressure at the top of chimney. It has been suggested to the writer that there are nodal points within the chimney owing to vibration of the chimney column and that this phenomenon may interfere with the results calculated from an elementary hypothesis.

47 The effective draft of a chimney will be reduced if ample area-ways are not provided for the inflow of air to the boiler room.

48 From the report of the Liquid Fuel Board, United States Navy, Bureau of Steam Engineering, during 1902 and 1903, the writer noted that in one particular test under forced draft conditions, where the weight of gases passing through the chimney was several times that corresponding to the natural draft conditions, giving rise to a discharge velocity of nearly 60 ft. per second, the calculated chimney friction was greater than 0.5 in. water pressure. The potential draft, due to the weight of chimney gases, was somewhat greater than 0.5 in., so that, neglecting the external influences, the draft at the base of chimney should have been practically negligible. On the other hand, the observed draft was almost as great as the potential draft, indicating a draft increment due to causes other than the difference in weight of the column of air inside and outside of chimney. On other tests the observed draft was considerably reduced, probably owing in part to the brushing aside by the wind of the hot column of gases immediately over the chimney.

49 An examination of the results of very elaborate tests made on a chimney at the Massachusetts Institute of Technology, giving draft observations at various portions, does not indicate a consistent relationship between the observed and calculated draft loss within the chimney for various points along the chimney shaft.

50 Sufficient tests have not been made to determine with accu-

racy the results from all of the influences noted, nor is it necessary that such tests be made. Chimneys must be designed with respect to the unfavorable surrounding conditions and with this idea in view it will be safe to ignore any increment in draft due to the extraordinary influences above cited.

51 Since the existing data in regard to the density of chimney gases refer particularly to coal-burning practice, it is necessary to consider possible variation with oil fuel, which is found to give a density of chimney gases somewhat greater than that with coal firing, varying with the excess air for combustion, the ultimate analysis of the crude oil, humidity of air for combustion, etc.

52 The specific volume of chimney gases is usually defined by

$$S = \frac{T}{K} \qquad [3]$$

in which

S = cu. ft. of gases per lb.

T = absolute temperature which, according to Marks and Davis = 459.64 + temperature, fahrenheit scale.

K = a constant

For dry air, standard barometer, K = 39.78.

53 For chimney gases at a temperature of 500 deg. fahr., 50 per cent excess of dry air being supplied for combustion, K = 39.73. In this instance the value of K is 0.13 of 1 per cent greater for dry air than for chimney gases, indicating a relative lessening in the weight of chimney column to this extent.

54 For 500 deg. fahr. temperature of chimney gases, 100 per cent excess of dry air supplied for combustion, K becomes 39.74, or but slightly heavier than for chimney gases with 50 per cent excess air.

55 For atmospheric air at 80 per cent humidity, K becomes 39.29, a decrease in density as compared with dry air of 1.2 per cent.

56 For chimney gases at a temperature of 500 deg. fahr., 50 per cent excess air for combustion at 80 per cent humidity, K becomes 39.38.

57 It is thus evident that the relative density of chimney gases is slightly greater than that of atmospheric air, considering the specified 50 per cent humidity, as against the relative decrease in density when considering dry air only. It follows that with 50 per cent excess air, 500 deg. temperature of chimney gases, there

is a degree of saturation of the atmosphere less than 80 per cent humidity, at which the relative density of chimney gases is the same as for atmospheric air.

58 For 500 deg. fahr. temperature of chimney gases, 100 per cent excess air at 80 per cent humidity, the value of K becomes 39.34, approaching still more nearly than for 50 per cent excess air the relative density of the atmospheric column.

59 In these instances the greatest relative difference in the relative densities of chimney column and atmospheric air is less than ¼ of 1 per cent, which for ordinary conditions would affect the calculated draft power of the chimney by less than ½ of 1 per cent. Since these differences are small compared with the uncertain value of the correcting factor f, as will be explained, and uncertain chimney calculations in general, it may be considered safe in calculations for chimney draft to ignore the difference in relative density at chimney column and atmospheric air, assuming throughout the same relative density for both and as equivalent to the atmospheric column based on dry air. A humid atmosphere will affect the performance of a boiler plant, tending slightly towards a sluggish combustion, and by reason of a slightly increased volume and velocity of gases, there is very small increase in the frictional resistances; these effects, however, may be considered of secondary importance.

60 The calculation of the potential draft of chimney is in accordance with

$$D = fKH \qquad [4]$$

in which

f = empirical coefficient depending largely on the rate of cooling gases within the chimney

K = theoretical draft per ft. of chimney due to difference in weight of atmospheric air and chimney gases, assuming the same relative density for both as for dry air at standard barometer, the chimney column being assumed throughout at the temperature of the entering gases

H = height in ft. of chimney above point of draft measurement

For altitudes other than sea level, the draft varies directly as the normal barometer.

61 Based on the results from actual practice from a number of unlined steel chimneys, the writer has assumed the value $f =$

0.95. This factor is subject to considerable variation. In chimneys having relatively large radiating surface, and low velocity of gases, the factor f may be 0.85 or lower. If the chimney be properly designed for a certain maximum load, then, with all boilers in operation, the velocity within the chimney will be normal, and the value of f should not vary greatly.

62 A German engineer makes the factor f dependent on the difference between the mean temperature inside the chimney and the mean temperature of external air, but this formula is still at fault in neglecting the chimney velocity and cooling area of chimney surface. It does not appear possible to define the factor f by any simple rational formula.

63 The value of K is determined on the conventional basis. It is important, however, to note that in determining K, when used in connection with the factor f, the temperature should be that of the gases entering the chimney and not the mean temperature of gases within chimney.

64 Table 2 gives the calculated potential draft of a chimney 100 ft. in height for various temperatures of gases entering the chimney and for various temperatures of external air. Column A shows the theoretical draft without loss due to cooling or friction; column B the theoretical draft corrected for a temperature drop in stack and density of flue gases, based on the value $f = 0.95$; column C the net draft based on the stated friction ratio as referred to below. It is important to note that the drafts are seldom measured at the floor line, and in computing chimney heights, allowance should be made for the height above floor line, of the point of draft measurement.

65 In order to obtain the draft within chimney at the point of entering gases, it is necessary to deduct from the values given in Table 1, Column B, the frictional loss within the chimney, which may be calculated

$$D = \frac{CW^2TH}{d^5} \qquad \ldots\ldots\ldots\ldots\ldots\ldots [5]$$

in which the nomenclature for D, T, W and H will apply as above. C will be represented by a coefficient from practice. For steel chimneys, the writer assumes $C = 2.3$. A German authority gives a frictional coefficient for brick chimneys which reduced to the same form would give $C = 2.5$.

66 The writer has not assumed differing values for steel, brick, and concrete chimneys. While manifestly a difference will

exist, it is believed that the increased friction loss due to masonry will be somewhat offset by the greater rate of cooling of an unlined steel chimney, and with the ordinary heights required for burning oil fuel, these differences will largely disappear.

67 The "Stirling" book recommends that, for brick chimneys, the diameter be made 6 per cent greater than for steel. A German engineering association as a result of its investigations recommended for steel chimneys that the diameter be nearly one-third larger than for brick chimneys.

68 The attendant difficulties in the exact determination of draft loss are so complicated by external influences, as already pointed out, that it is probable that exact information will be long delayed.

69 Based on equation [2] for the resistance of a right angle turn and equation [1] for the resistance of a flue, a simple calculation will show that the loss due to velocity head or the loss due to a right angle turn at flue velocity is equivalent to the resistance of a length of steel flue equal to 10.4 diameters.

70 Based on equation [5] the loss due to velocity head or a right angle turn at chimney velocity will be equivalent to the resistance of a height of steel chimney equal to 13.6 diameters.

TEMPERATURE OF GASES

71 As is well known, the temperature of gases will depend upon the type of boiler, rate of firing boiler, rate of excess air supply, and general condition of boilers as affecting thorough circulation of gases and absorption of heat.

72 In economical types of boilers under test conditions, escaping temperatures are at rating about 425 deg. fahr., or slightly under, and at 50 per cent overload, 500 deg. fahr. or slightly under. With economical boilers under good firing, with tube surfaces in good condition, the boiler temperature should be about 450 deg. fahr, or under, at rating, and about 525 deg. fahr. or under, at 50 per cent overload.

73 It is necessary to allow for a drop in temperature between the boiler outlet and the chimney inlet, depending upon the length and character of the breeching, air infiltration, etc.; also upon the number of idle boilers connected to the same breeching, which may be passing cold air through dampers and which ordinarily are not close fitting.

74 In Table 5 for central chimneys, the writer has

assumed initial temperature of chimney gases 425 deg. fahr. for boilers operating at rating, and 500 deg. fahr. for boilers operating at 50 per cent overload. For chimneys directly mounted over boilers and with short breeching the relative temperature entering the chimney will be greater than indicated. Under unfavorable firing conditions and with less economical types of boilers the temperatures are materially greater than indicated.

ATMOSPHERIC TEMPERATURE

75 As is well known, but frequently overlooked, the temperature of the atmospheric column influences materially the potential draft of a chimney. By referring to Tables 2 and 3 the variation in draft, and correspondingly the variations necessary in chimney height due to range of atmospheric temperature, is appreciable. During cold weather record capacities of chimney are obtained which will not hold for extreme summer heat; the atmospheric temperature may exceed 100 deg. fahr., and in certain locations, 120 deg. fahr. The location of the boiler plant will of necessity decide the consideration given to this factor.

GENERAL FORMULÆ

76 Consideration will here be given to three types of chimneys, differing in respect to the amount of draft lost between the boiler and chimney, as follows:

77 *Case I: Independent chimney for each boiler, mounted directly on boiler, top outlet smoke connection.* In considering this case, the writer has in mind the use of Stirling boilers where an A-shaped breeching is usually employed to connect from the top flue outlet converging to the chimney diameter along lines which involve little frictional resistance within the breeching. In such cases the breeching may ordinarily be considered as the equivalent of the same height of chimney, and a separate loss due to breeching entirely ignored. Where a battery of boilers is connected to one chimney with an A-shaped overhead breeching, the case is but slightly different.

78 *Case II: Independent chimneys direct-connected to a single boiler, or a battery of two boilers, the chimney being located in the rear of boiler or boilers, a short horizontal flue being used.* In this instance the resistance of the breeching itself is almost negligible, but there is added, as compared with Case I, a greater

TABLE 2 CHIMNEY DRAFT—OIL FUEL. SHOWING THEORETICAL AND AVAILABLE DRAFT IN CHIMNEY 100 FT. IN HEIGHT ABOVE THE POINT OF DRAFT MEASUREMENT FOR VARIOUS TEMPERATURES AT SEA LEVEL

Temperature of Gases entering Chimney, Deg. Fahr.	Chimney Draft, In. of Water								
	Temperature of Outside Air = 60 Deg. Fahr.			Temperature of Outside Air = 80 Deg. Fahr.			Temperature of Outside Air = 100 Deg. Fahr.		
	A	*B*	*C*	*A*	*B*	*C*	*A*	*B*	*C*
280	0.436	0.414	0.345	0.382	0.363	0.303	0.332	0.315	0.263
300	0.464	0.441	0.367	0.410	0.390	0.325	0.359	0.341	0.284
320	0.490	0.466	0.388	0.435	0.413	0.344	0.385	0.366	0.305
340	0.514	0.488	0.407	0.460	0.437	0.364	0.409	0.389	0.324
360	0.537	0.511	0.425	0.483	0.459	0.383	0.432	0.411	0.342
380	0.560	0.532	0.444	0.505	0.480	0.400	0.455	0.432	0.360
400	0.581	0.550	0.460	0.527	0.501	0.417	0.476	0.452	0.377
425	0.606	0.576	0.480	0.552	0.524	0.437	0.501	0.476	0.397
450	0.630	0.598	0.499	0.576	0.547	0.456	0.525	0.498	0.416
475	0.652	0.619	0.516	0.598	0.568	0.475	0.547	0.520	0.433
500	0.674	0.640	0.534	0.619	0.588	0.490	0.569	0.541	0.451
525	0.694	0.659	0.550	0.639	0.607	0.506	0.589	0.560	0.466
550	0.713	0.678	0.565	0.659	0.627	0.522	0.608	0.576	0.482
575	0.731	0.694	0.579	0.677	0.643	0.536	0.626	0.595	0.496
600	0.749	0.712	0.593	0.694	0.657	0.550	0.644	0.612	0.510
625	0.765	0.727	0.606	0.711	0.675	0.563	0.660	0.627	0.523
650	0.781	0.742	0.619	0.727	0.691	0.576	0.676	0.642	0.535
675	0.796	0.756	0.630	0.742	0.705	0.588	0.691	0.656	0.547
700	0.811	0.768	0.642	0.756	0.718	0.599	0.706	0.671	0.559
725	0 825	0.783	0.653	0.770	0.731	0.610	0.720	0.684	0.570
750	0.838	0.797	0.664	0.783	0.744	0.620	0.733	0.697	0.581

Column A shows the theoretical draft due to a height of chimney of 100 ft. and $=KH$ where: H = height in feet of chimney above point of draft measurement = 100.

K = theoretical draft per foot of chimney due to difference in weight of atmospheric air and chimney gases at stated temperatures, assuming same relative density for both as for dry air = one one-hundredth of value given in column *A*.

Column B shows draft in *A* corrected for temperature drop in stack and density of flue gases, $B = f\,A = f\,KH. f$ is variable, but assumed above as 0.95.

Column C shows net draft, being draft in column *B* corrected for friction, assumed as $\frac{5}{6}$ of draft in column *B*.

Chimney friction should be $\frac{1}{6}$ total draft for minimum first cost of chimney when mounted directly over boiler or battery of two boilers. This does not apply to horizontal breeching or to central chimneys combining several losses.

For altitude draft varies directly as normal barometer.

TABLE 3 APPROXIMATE MAXIMUM CAPACITY; ACTUAL BOILER HORSEPOWER; STEEL CHIMNEYS; SIZES FOR MINIMUM COST OF CHIMNEY; INDEPENDENT TYPE DIRECTLY MOUNTED OVER SINGLE OR DOUBLE BOILER SETTING WITH TOP OUTLET; 50 PER CENT EXCESS AIR SUPPLY; SEA LEVEL. OIL FUEL.

Temp. of Gases entering Chimney, Deg. Fahr.		300	400	500	600	700	Net Draft required at Point of Measurement In. of Water	Height of Chimney in Ft. above Point of Draft Measurement, for Various Temperatures of Entering Gases and Outside Air, corresponding to Tabulated Draft														
Assumed Boiler Efficiency		80.4	77.1	73.8	70.5	67.3																
Lb. of Chimney Gases per B.h.p.		49.5	51.6	54.0	56.3	59.0		300 Deg.			400 Deg.			500 Deg.			600 Deg.			700 Deg.		
Diam., In.	Area, Sq. Ft.	Actual Boiler Horsepower Temp. Outside Air, 80 Deg. Fahr.						Temperature of Air, Deg. Fahr.														
								60	80	100	60	80	100	60	80	100	60	80	100	60	80	100
18	1.77	61	62	61	59	56	0.01	2.7	3.1	3.5	2.2	2.4	2.7	1.9	2.0	2.2	1.7	1.8	2.0	1.6	1.7	1.8
21	2.41	90	92	90	87	82	0.02	5.5	6.2	7.0	4.3	4.8	5.3	3.7	4.1	4.4	3.4	3.7	3.9	3.1	3.3	3.6
24	3.14	125	127	125	121	115	0.03	8.2	9.2	10.6	6.5	7.2	8.0	5.6	6.1	6.7	5.1	5.5	5.9	4.7	5.0	5.4
27	3.98	168	172	168	163	154	0.04	10.9	12.3	14.1	8.7	9.6	10.6	7.5	8.2	8.9	6.8	7.3	7.8	6.2	6.7	7.2
30	4.91	219	223	219	212	201	0.05	13.7	15.4	17.6	10.9	12.0	13.3	9.4	10.2	11.1	8.5	9.2	9.8	7.8	8.4	9.0
33	5.94	278	283	278	268	255	0.06	16.3	18.5	21.1	13.0	14.4	15.9	11.2	12.2	13.3	10.1	11.0	11.8	9.4	10.0	10.7
36	7.07	345	352	345	334	317	0.07	19.1	21.6	24.7	15.2	16.8	18.6	13.1	14.3	15.5	11.8	12.8	13.7	10.9	11.7	12.5
39	8.30	422	430	422	408	387	0.08	21.8	24.6	28.1	17.4	19.2	21.2	15.0	16.3	17.8	13.5	14.6	15.7	12.5	13.4	14.3
42	9.62	508	518	508	490	465	0.09	24.6	27.7	31.7	19.5	21.6	23.4	16.8	18.4	20.0	15.2	16.5	17.6	14.0	15.0	16.1
45	11.05	602	615	602	583	553	0.10	27.3	30.8	35.2	21.7	24.0	26.5	18.7	20.4	22.2	16.9	18.3	19.6	15.6	16.7	17.9
48	12.57	708	723	708	685	650	0.20	54.5	61.6	70.4	43.4	48.0	53.0	37.4	40.8	44.4	33.8	36.6	39.2	31.2	33.4	35.8
54	15.90	951	971	951	919	872	0.30	81.9	92.4	106	65.1	72.0	79.5	56.1	61.2	66.6	50.7	54.9	58.8	46.8	50.1	53.7
60	19.64	1240	1260	1240	1197	1134	0.40	109	123	141	86.8	96.0	106	74.8	81.6	88.8	67.6	73.2	78.4	62.4	66.8	71.6
66	23.76	1570	1600	1570	1520	1440	0.50	137	154	176	109	120	133	93.5	102	111	84.5	91.5	98.0	78.0	83.5	89.5
72	28.27	1950	1990	1950	1890	1790	0.60	163	185	211	130	144	159	112	122	133	101	110	118	93.6	100	107

TABLE 3—*Continued*

78	33.18	2390	2430	2390	2310	2190	0.70	191	216	247	152	168	186	131	143	155	118	128	137	109	117	125
84	38.49	2870	2930	2870	2770	2630	0.80	218	246	281	174	192	212	150	163	178	135	146	157	125	134	143
90	44.18	3410	3480	3410	3300	3130	0.90	246	277	317	195	216	234	168	184	200	152	165	176	140	150	161
96	50.27	4010	4090	4010	3870	3670	1.00	273	308	352	217	240	265	187	204	222	169	183	196	156	167	179

Conversion Factors for 60 Deg. and 100 Deg. Atmosphere

Temp. of Gases	300	400	500	600	700
Air at 60 Deg.	1.06	1.05	1.04	1.04	1.04
Air at 100 Deg.	0.935	0.948	0.960	0.965	0.970

This table indicates diameter of chimney for minimum first cost of chimney when cost is proportional to the product of diameter and height. While capacities apply to any arrangement of chimney, cost is minimum only when gases flow vertically into chimney without horizontal breeching and without loss due to right angle turn of gases into chimney. Diameter for minimum cost is independent of draft resistance, if height be selected from the table corresponding to total draft resistance of boiler and damper outlet. For total chimney height above floor line, add to stated heights the actual height above floor line to point of draft measurement. For stated condition of minimum chimney cost, draft resistance is equal to one-sixth of static draft and this allowance has been made in table. See Table 1 for basis of assumed boiler efficiencies, and Table 2 for basis of draft calculations. If boilers have superheaters increase b.h.p. to include heat absorbed by superheater.

draft loss in the damper outlet and turn into flue, also the loss due to a right angle turn of the gases on entering chimney.

79 *Case III: A central chimney connected to a battery of two or more boilers by means of a common flue of appreciable length.* In the computation for this case, the writer will assume the prevailing practice in Babcock and Wilcox installations, the damper and flue outlet being directly in the rear and there being a straight run of flue to chimney. In this instance there is involved not only the loss through the damper outlet, but the loss due to the turn from the damper outlet into the breeching, loss due to friction of the breeching, and also that due to the right angle turn from the breeching into the chimney. A modification of this arrangement would include installations where there is a resistance due to turns of the breeching in which instances losses may be minimized by avoiding sharp bends.

80 The fundamental formula, ignoring economizers for the present, is written by equating the potential draft of chimney and the sum of various resistances

$$fKH = D_1 + D_2 + D_3 + D_4 + D_5 + D_6 + D_7 \ldots\ldots [6]$$

in which

f, K, H are as heretofore stated

D_1 = draft resistance of boiler measured on boiler side of damper

D_2 = draft resistance due to damper box

D_3 = draft resistance due to right angle turn into breeching

D_4 = draft resistance due to straight length of breeching

D_5 = draft resistance due to turns in breeching

D_6 =— draft resistance due to turn from breeching into chimney

D_7 = draft resistance of chimney calculated in accordance with [5]

81 D_1 and D_2 usually depend on the boiler in question, and required load or overload. Since the flue area is usually fixed by the chimney area, the values of D_3, D_4, D_5, D_6 and D_7 are interrelated as more fully explained below.

82 When investigating an existing plant under assumed load conditions, it is merely necessary to calculate the various load resistances, comparing their total with that which the chimney is capable of producing.

83 For Case I, D_3, D_4, D_5 and D_6 disappear, and the equation becomes

$$fKH = D_1 + D_2 + \frac{2.3\ W^2\ HT}{d^5} \qquad [7]$$

from which

$$\text{B.h.p.} = \frac{60^2\ \sqrt{d^5}}{G} \sqrt{\frac{fKH - (D_1 + D_2)}{2.3\ HT}} \qquad [8]$$

in which

$G =$ lb. of chimney gases per b.h.p. per hour

84 Table 3 gives chimney capacities for Case I based on selection of height and diameter for minimum first cost of chimney as explained below.

85 The treatment for Case II is simple, after considering the more general Case III. For the latter it will be assumed that all flues are rectangular in shape, having a depth of 1.5 times the width, giving a flue resistance 15 per cent in excess of that for a circular flue of equal area and length, which length of flue is ten times the chimney diameter; that flue area is maintained substantially its entire length; and that owing to the decreasing velocities at increasing distances from chimney, the actual resistance of the straight run of breeching will be practically equivalent to that having uniform velocity its entire length, but 5 chimney diameters in length instead of 10 chimney diameters its actual length. D_1 will be as found from test. D_2 and D_3 cannot easily be separated and will be considered as D_2 properly evaluated from test. Then

$$D_4 = \frac{2.3\ W^2 T \left(\frac{5d}{12}\right)}{d^5} \qquad [9]$$

D_5 disappears from Case III; but when breeching contains a right angle turn, D_5 may be taken as equivalent to the resistance of chimney height of 9.4 chimney diameters. Then

$$D_6 = \frac{2.3\ W^2 T \left(\frac{13.6d}{12}\right)}{d^5} \qquad [10]$$

Equation [6] may now be written

$$fKH = (D_1 + D_2) + \frac{2.3\ W^2 T}{d^5}\left(H + \frac{18.6d}{12}\right) \qquad [11]$$

The coefficient $\frac{18.6}{12} = 1.55$; in the table for central chimneys this term, for simplicity of calculation, has been taken as 1.5.

86 As equation [11] involves terms of the fourth and fifth powers, it is simpler to calculate the boiler horsepower of chimney from assumed diameter and height, when [11] finally becomes

$$\text{B.h.p.} = \frac{60^2}{G} \sqrt{d^5} \sqrt{\frac{fKH - (D_1 + D_2)}{2.3\ T\ (H + 1.5d)}} \ldots\ldots\ldots\ldots [12]$$

87 In computing the table of capacity for central chimneys, Case III, $D_1 + D_2$ has been taken as 0.15 in. for boilers at rating and 0.3 in. for boilers for 50 per cent overload, which values are conservative for standard boilers, 12 tubes high; for 9 high boilers the height may be slightly reduced, and for 14 high boilers the height should be increased in keeping with draft resistance data above given. Table 4 gives calculated capacities for Case III based on various other assumed data given in tables and explained in the footnotes.

88 For Case II, in which the resistance of straight run of breeching is neglected, the coefficient of d, 1.5, in the term $(H + 1.5\ d)$, equation [12] becomes instead, $\frac{13.6}{12}$ or 1.13; where one chimney is connected to two boilers symmetrically arranged, there is a right angle turn in breeching, as well as into chimney, and the above coefficient becomes $\frac{13.6 + 9.4}{12}$ or 1.92 approximately.

89 For Case III, considering one right angle turn in breeching, the coefficient of d becomes 2.33.

90 In the foregoing it is assumed, as stated above, that the total flue area is 20 per cent greater than chimney area.

91 A study of the term $(H + 1.5\ d)$, equation [12], shows that for the conditions assumed, the resistance of flue and turn into chimney, both dependent on chimney diameter, is of as much influence on chimney performance as the resistance of chimney itself, whenever the height of chimney above the point of draft measurement is less than 12×1.5 or 18 diameters. Thus, in a 5-ft. chimney the above-named friction losses equal the chimney friction when chimney height above point of draft measurement is 90 ft. or less.

CHIMNEY PROPORTIONS FOR MINIMUM FIRST COST OF CHIMNEY

92 It is obvious that an indefinite increase in the diameter of chimney will not decrease the height below that necessary to maintain the draft corresponding to the fixed boiler resistances. An increase in height will reduce the diameter only to the point where the added draft per foot of height equals the chimney resistance per foot of height. Between these two extremes there are numberless combinations of height and diameter which will maintain the desired draft at base of chimney or the rear of boiler. But for a given set of conditions and for the type of construction under consideration there is but one height and one diameter resulting in a chimney of least first cost.

93 The simplest case is that for steel chimneys having a given average thickness of material, the cost of chimney being proportional to the product of height and chimney diameter.

94 For Case I, the final equation for minimum first cost, derived by the method of differential calculus, is reducible to simplest form. A chimney of least first cost, for the simple case stated, for a constant maintained draft inside chimney at chimney base, is that in which the frictional resistance of chimney is one-sixth of the potential draft of chimney.

95 This relationship was first stated in the book "Stirling," but incorrectly, as regards central chimneys as well as direct mounted chimneys, as in Case I. For Cases II and III this relationship will not apply, and no simple formula can be derived for these cases, owing to the resistance of breeching and chimney inlet, the latter involving terms of the fourth power. Resistance of chimney and breeching introduces terms of the fifth power, the final equation of minimum cost being too complex for ordinary engineering purposes. Equating the chimney resistance and one-sixth of potential draft of chimney we have

$$\frac{1}{6} fKH = \frac{2.3\ W^2\ TH}{d^5} \quad \ldots\ldots\ldots\ldots\ldots\ldots [13]$$

It is apparent that the term H disappears, simplifying [13] to

$$\text{B.h.p.} = \frac{60^2}{G}\ \sqrt{d^5}\ \sqrt{\frac{\frac{1}{6} fK}{2.3\ T}} \quad \ldots\ldots\ldots\ldots\ldots [14]$$

96 The draft at base of chimney becomes, by the above relationship, five-sixths of potential draft of chimney. Column C, Table 1, gives chimney drafts on the above basis. Table 3 gives

TABLE 4—PART 1 APPROXIMATE MAXIMUM CAPACITY; ACTUAL BOILER HORSEPOWER; STEEL CHIMNEYS WITH SHORT FLUES; CHIMNEYS CENTRALLY LOCATED; STATIONARY B. & W. BOILERS.

Based on Atmospheric Temperature of 80 deg. fahr.; Sea Level, and Other Assumed Data. Oil Fuel

Ratio of actual boiler h.p. to rated boiler h.p., per cent	100
Draft in inches of water at boiler outlet, chimney side of damper	0.15
Corresponding excess air through boiler, per cent	50
Assumed excess air supply for determining boiler efficiency, chimney diameter and draft resistance of chimney and breeching, per cent	50
Assumed temperature of gases leaving boiler, deg. fahr.	450
Assumed temperature of gases entering chimney, deg. fahr.	425
Assumed boiler efficiency, working, not test conditions, per cent	75.4
Assumed lb. of chimney gases per actual boiler h.p.	52.8

Diameter, In.	Area, Sq. Ft.	Height in Feet above Boiler Room Floor Line											
		50	60	70	80	90	100	110	120	130	140	150	160
18	1.77	57	77	88	97	103	108	112	115	118	120	122	124
21	2.41	81	109	127	139	149	156	162	167	171	175	178	180
24	3.14	110	149	173	190	204	214	222	229	235	240	245	249
27	3.98	144	194	226	250	268	282	293	303	311	318	325	330
30	4.91	182	247	288	318	342	361	376	388	400	409	417	424
33	5.94	224	306	358	396	427	450	470	486	500	513	523	533
36	7.07	272	372	436	483	521	550	575	596	614	629	642	655
39	8.30	325	444	521	582	626	662	692	718	740	758	776	790
42	9.62	382	522	615	686	741	785	821	852	880	902	923	940
45	11.05	444	610	719	801	869	920	961	1000	1030	1060	1080	1110
48	12.57	512	702	830	927	1000	1060	1120	1160	1200	1230	1260	1290
54	15.90	660	911	1080	1210	1310	1390	1460	1520	1570	1610	1660	1700
60	19.64	827	1150	1360	1520	1660	1760	1860	1940	2000	2060	2110	2160
66	23.76	1020	1410	1670	1880	2050	2180	2300	2400	2480	2560	2630	2690
72	28.27	1220	1700	2020	2280	2490	2650	2790	2910	3030	3130	3210	3290
78	33.18	1450	2020	2400	2710	2970	3170	3340	3490	3620	3750	3850	3950
84	38.49	1690	2370	2830	3190	3490	3730	3940	4130	4280	4430	4560	4670
90	44.18	1960	2740	3280	3710	4060	4350	4600	4810	5000	5170	5320	5460
96	50.27	2250	3150	3770	4260	4670	5000	5300	5560	5780	5980	6160	6320
102	56.75	2560	3580	4290	4850	5320	5710	6050	6340	6610	6850	7060	7250
108	63.62	2880	4030	4840	5480	6030	6470	6850	7180	7500	7770	8010	8230
114	70.88	3220	4510	5440	6170	6780	7280	7720	8100	8440	8750	9050	9300
120	78.54	3580	5030	6060	6870	7560	8130	8630	9060	9440	9800	10100	10400
126	86.59	3960	5580	6710	7630	8380	9030	9590	10100	10510	10900	11300	11600
132	95.03	4390	6150	7420	8420	9260	9990	10600	11200	11700	12100	12500	12900
138	103.9	4800	6730	8150	9270	10200	11000	11700	12300	12800	13300	13800	14200
144	113.1	5230	7370	8920	10200	11200	12000	12800	13500	14100	14600	15100	15600
156	132.7	6210	8730	10600	12000	13200	14300	15200	16000	16700	17400	18000	18600
168	153.9	7190	10200	12300	14000	15500	16800	17800	18800	19700	20500	21200	21900
180	176.7	8350	11800	14300	16200	17900	19400	20700	21800	22900	23800	24700	25400

Effective height of chimney or height above point of draft measurement assumed 12 ft. less than tabulated heights corresponding to 12 high B. & W. boilers, 7 ft. furnace height. Draft resistance of boiler correct for 12 high boilers: capacities somewhat less for 14 high and greater for 9 high. Capacity reduced by long breeching. For correction for altitude, Table 5 or draft calculations, Table 2: for boiler efficiencies, Table 1; for capacity independent chimneys, able 3. When building conditions permit, select chimneys of least height in table for minimum

TABLE 4—PART 2 APPROXIMATE MAXIMUM CAPACITY; ACTUAL BOILER HORSEPOWER; STEEL CHIMNEYS WITH SHORT FLUES; CHIMNEYS CENTRALLY LOCATED; STATIONARY B. & W. BOILERS

BASED ON ATMOSPHERIC TEMPERATURE OF 80 DEG. FAHR.; SEA LEVEL, AND OTHER ASSUMED DATA. OIL FUEL

Ratio of actual boiler h.p. to rated boiler h.p., per cent.................................. 150
Draft in inches of water at boiler outlet, chimney side of damper.......................... 0.30
Corresponding excess air through boiler, per cent..................................Less than 50.
Assumed excess air supply for determining boiler efficiency, chimney diameter and draft resistance of chimney and breeching, per cent.................................. 50
Assumed temperature of gases leaving boiler, deg. fahr.................................. 525
Assumed temperature of gases entering chimney, deg. fahr.................................. 500
Assumed boiler efficiency, working, not test conditions, per cent.......................... 73.0
Assumed lb. of chimney gases per actual boiler h.p.................................. 54.6

Diameter, In.	Area. Sq. Ft.	Height in Feet above Boiler Room Floor Line								
		80	90	100	110	120	130	140	150	160
18	1.77	63	75	84	91	96	101	104	108	110
21	2.41	90	108	121	131	139	146	151	156	160
24	3.14	123	148	166	180	191	201	208	215	221
27	3.98	161	195	219	238	253	265	276	285	293
30	4.91	206	249	280	304	324	340	354	366	377
33	5.94	256	310	349	381	405	426	444	459	472
36	7.07	312	379	427	466	497	523	545	564	581
39	8.30	376	455	514	561	599	631	657	681	701
42	9.62	443	539	609	665	711	749	782	810	835
45	11.05	518	630	713	779	834	879	918	952	981
48	12.57	599	729	827	904	967	1020	1070	1110	1140
54	15.90	779	951	1080	1180	1270	1340	1400	1460	1500
60	19.64	985	1200	1370	1500	1610	1710	1790	1860	1920
66	23.76	1220	1490	1700	1860	2000	2120	2220	2310	2390
72	28.27	1470	1810	2060	2260	2430	2580	2710	2820	2910
78	33.18	1750	2150	2460	2710	2910	3090	3250	3380	3500
84	38.49	2060	2530	2900	3190	3440	3650	3840	4000	4150
90	44.18	2390	2950	3370	3720	4010	4260	4480	4670	4850
96	50.27	2750	3390	3880	4290	4630	4920	5180	5400	5610
102	56.75	3140	3870	4440	4900	5290	5630	5930	6190	6430
108	63.62	3550	4380	5020	5550	6000	6390	6730	7030	7300
114	70.88	3990	4920	5650	6250	6760	7200	7590	7930	8250
120	78.54	4440	5490	6310	6990	7560	8060	8490	8890	9240
126	86.59	4930	6100	7020	7770	8410	8970	9460	9900	10300
132	95.03	5450	6740	7760	8600	9310	9930	10500	11000	11400
138	103.9	5990	7420	8530	9460	10300	10900	11600	12000	12600
144	113.1	6550	8120	9350	10400	11200	12000	12700	13300	13800
156	132.7	7760	9630	11100	12300	13400	14300	15100	15800	16500
168	153.9	9060	11300	13000	14400	15700	16800	17700	18600	19400
180	176.7	10500	13000	15100	16700	18200	19500	20600	21600	22600

cost of chimney. Maximum chimney capacities stated are for uniform continuous load equally divided all boilers. For large plants subject to hand regulation, or for swinging load, reduce capacity from 10 to 20 per cent. If boilers have superheaters increase b.h.p. to include heat absorbed by superheater. Breeching 20 per cent excess stack area, without turns, length not exceeding 10 chimney diameters; boilers uniformly distributed along breeching; breeching depth = 1.5 × width. For capacities of chimneys with economizers, see Table 7. For economizer draft resistance, see Table 6.

capacities of economical chimneys for various assumed conditions, also heights for various required drafts. It will be noted that diameter is dependent only on boiler horsepower, and height dependent only on draft resistance of the boiler in question, information which must be supplied by the designer. Table 3 indicates chimneys of relatively large diameter but moderate height. Whenever conditions compel greater chimney height than given in table, the diameter may be reduced to maintain the required draft at chimney base, in accordance with equation [7].

97 The low chimneys indicated by Table 3 are noteworthy. For example: Assuming a Stirling boiler, requiring 0.1 in. draft at rating, with 450 deg. temperature, draft measurement 15 ft. above floor line, for 60 deg. atmosphere the height of chimney by interpolation is about 20 ft., to which must be added the 15 ft., making a total height of 35 ft. above floor. Under test conditions the draft in certain sizes is as low as 0.05 in., for which the height would be 11 ft. plus 15 ft. or 26 ft. above floor. At 100 per cent overload this total height would be increased to from 50 to 100 ft., depending on assumed draft resistance of boiler, atmospheric temperature, etc.

98 In general, it may be said that for Cases II and III the chimney of least first cost will be of moderate height, although this height will of necessity be somewhat greater than indicated by Table 3 for Case I. The least height in Table 4 corresponds to minimum first cost for the range of height intervals chosen. A cut-and-try method must be resorted to for greater refinement.

REDUCTION OF STACK CAPACITY AT ALTITUDES

99 The writer is not aware of any treatise dealing correctly with stack capacities at altitudes, either for coal or for oil fuel. Heretofore it has been assumed that the stack height increases inversely as the barometric pressure and that the stack diameter increases inversely as the two-fifths power of the barometric pressure, all other boiler and chimney data remaining constant. This relationship is based on the assumption of constant draft, measured in inches of water at the base of chimney for a given rate of firing of boilers, either expressed in lb. of coal per sq. ft. of grate surface or in ratio of rated capacity when burning fuel oil, regardless of altitude, an assumption wholly incorrect.

100 Whenever a chimney is working at maximum capacity

either at sea level or at an altitude, it is evident that the entire potential draft is utilized in overcoming the various resistances, each proportional to the square of the velocity, which varies in different points in the path of the gases.

101 Owing to the fixed areas along the path, all velocities may be related to a common velocity, say that within the chimney, and therefore all resistances in a given boiler plant for a given temperature of gases, etc., may be expressed as proportional to the square of the chimney velocity.

102 The total resistance to flow in terms of velocity head may be expressed in terms of the weight of a column of external air, and the numerical value of this head will be independent of the barometric pressure. Likewise the potential draft of a chimney, expressed in height of column of external air, will be numerically independent of the barometric pressure.

103 It is therefore evident that if a given boiler plant with connected chimney, operated with a fixed fuel, be transplanted from sea level to an altitude and if the temperatures remain constant (an assumption not entirely correct), the total draft head, measured in height of column of external air, will be numerically constant. The velocity of chimney gases will therefore remain the same at altitude as at sea level, and the weight of gases flowing per second with fixed velocity will be proportional to the atmospheric density, or inversely proportional to the normal barometric pressure.

104 Hence, except for errors due to assumption of constant temperatures of gases, it may be stated that the maximum capacity of a given boiler plant with a fixed size of chimney and fixed fuel will decrease at altitudes higher than sea level at a rate directly proportional to barometric height. For example: At an altitude of approximately 5764 ft., the normal barometric pressure would be 80 per cent of the barometer at sea level, and a chimney connected to a given plant, giving a maximum capacity of 1000 b.h.p. at sea level, would give not more than 800 b.h.p. at this altitude. However, corresponding to the reduction in rate of load on boilers, there would be a decrease in the temperature of boiler and chimney gases, tending under normal conditions to a still further reduction in maximum capacity.

105 An error commonly made in the consideration of stack capacities at altitudes is to assume that a given grade of fuel at a fixed boiler rating will require at an altitude the same draft,

measured in inches of water, at boiler damper as at sea level. It is evident that to develop a given boiler horsepower requires a constant weight of chimney gases and air for combustion. Hence as the altitude is increased the density is decreased and, correspondingly, the velocity through the furnace, the bed of coal, or the firebrick checkerwork and the boiler passes, must be greater at altitude than at sea level. The mean velocity, therefore, for a given boiler horsepower and constant weight of gases will be inversely proportional to the barometric pressure and the velocity head, measured in column of external air, will be inversely proportional to the square of the barometric pressure.

106 It is possible and usually desirable in coal burning to select a differently arranged boiler for altitudes, particularly in the West, increasing the relative grate surface not only because of reduction of draft loss, but also by reason of the poorer grades of coal, excessive ash content, etc. These considerations, however, do not apply with oil fuel, although it is possible but not desirable to modify the arrangement of the boiler, reducing somewhat its draft resistance.

107 A given chimney with gases at a fixed velocity, temperature, etc., will result in a draft head, measured in height of column of external air, numerically independent of the barometer. The absolute draft pressure, measured in inches of water, will therefore be inversely proportional to barometric pressure.

108 It follows that for chimneys operating at altitude, it is necessary not only to increase the height, but also the chimney diameter. Corresponding to the increase in height, there is an added resistance within the chimney owing to the friction of the additional height. This frictional loss can be compensated by a suitable increase in the diameter, and when so compensated it is evident that the chimney height must be increased at a ratio inversely proportional to the square of the normal barometric pressure.

109 For Case I in which the chimney is mounted directly on top of the boiler, the gases entering the chimney without right angle turn, a simple calculation, based on equation [8], will indicate that the diameter of the chimney must be increased inversely as the two-fifths power of the barometric column. For example, for a height of 5764 ft. the normal barometer would be 80 per cent of that at sea level, and the height increases in the ratio of

1.5625 instead of 1.25 as is commonly assumed; the ratio of diameters will be 1.09.

110 If a chimney having height and diameter for minimum first cost be selected for sea level corresponding to a certain rating of boilers, then for Case I the modification indicated will still result in the most economical chimney for operation at altitude.

111 For Cases II and III the effective chimney height may still be increased in inverse proportion to the square of the normal barometer, and the diameter of chimney and breeching must also be increased in such ratio that the modified chimney and breeching will have the same frictional resistance at sea level as before increasing the chimney height at sea level. In other words, the coefficient of resistance of chimney and breeching must remain a constant. In the general Case III it was necessary to assume a relationship between flue length and chimney diameter; however, with the same boiler plant transplanted to an altitude, there is not, of necessity, any change in flue length. Therefore, if length and diameter of flue remain unchanged at altitude its coefficient of resistance remains constant.

112 If the chimney diameter be increased at altitude in the ratio set forth for Case I and the breeching be increased to 20 per cent excess area, as for sea level, then with constant length of breeching the coefficient of resistance of both breeching and chimney inlet actually decreases, indicating that by applying to Cases II and III the same correcting factor as for Case I, the modified chimney will not only be ample, but slightly larger than necessary.

113 The general formulæ for altitudes are derived as follows: Let

P_o = normal barometric pressure at sea level
P_a = normal barometric pressure at altitude, and
$R = \frac{P_o}{P_a}$

Then if K, D_1, D_2, etc., still indicate values for sea level, and H and d new values at altitude, equation [7] becomes

$$\frac{fKH}{R} = R\left(D_1 + D_2 + \frac{2.3\ W^2TH}{d^5}\right) \ldots\ldots\ldots\ldots [15]$$

also equation [8] becomes

$$\text{B.h.p.}=\frac{60^2}{G}\sqrt{d^5}\,\frac{\sqrt{\frac{fKH}{R^2}-(D_1+D_2)}}{\sqrt{2.3\,HT}}\ldots\ldots\ldots\ldots[16]$$

Likewise equation [12] becomes

$$\text{B.h.p.}=\frac{60^2}{G}\sqrt{d^5}\,\frac{\sqrt{\frac{fKH}{R^2}-(D_1+D_2)}}{\sqrt{2.3T\,(H+1.5d)}}\ldots\ldots\ldots\ldots[17[$$

114 Equation [17] assumes breeching length 10 chimney diameters, which is not necessarily the case, as pointed out above.

115 It is apparent that T is not modified by altitude, with the same rate of load on boilers and the same excess air.

116 Table 5 gives correcting factors for various altitudes.

MODIFICATION OF CHIMNEY CAPACITY DUE TO FUEL ECONOMIZERS

117 Owing to the comparatively small draft required when burning crude oil, it is unnecessary to resort to mechanical draft when installing fuel economizers.

118 It is self-evident that if economizers be added to an existing boiler plant, the available draft at boiler outlet will be decreased, not only by reason of the added frictional resistance of fuel economizers, but also because of the lowering of the temperature of gases within the chimney. It is also clear that the chimney dimensions must be modified to counteract these influences.

119 For the purpose of this paper, the writer has assumed the economizer dimensions as given in the Green Fuel Economizer Company's catalogue. In determining the draft resistance of economizers there is involved not only the possible increased number of flue turns due to the installation of economizer, but also the frictional resistance of economizer tubes.

120 The writer has assumed that an economizer should be installed with an ample area for the flow of gases with the economizer "side dampers" closed, and the resistance of the economizers has been determined on this basis; the opening of side dampers then provides latitude for increased draft at boiler.

121 The resistance data for economizers indicates a lack of concordance, doubtless due to the fact that draft observations are usually made incidental to boiler and economizer tests and not for the specific purpose of noting the draft resistance of economizers. Owing to error of readings, the draft resistance of econo-

mizer being estimated as a difference of two draft observations, it is possible that the errors may be in opposite directions, tending to increase the actual draft resistance, or vice versa. Draft readings may be influenced by the velocity of flow of gases or by the relative heights of the two points of observation.

122 Gas analyses are usually taken at the boiler and not in the economizer setting, so that tests of record do not indicate closely the degree of excess air passing through economizer or the influence of air leakage in flues and economizer setting.

TABLE 5 CORRECTION FOR ALTITUDE; CHIMNEY CAPACITIES. OIL FUEL

Altitude, Feet above Sea Level	Normal Barometer	R Ratio Barometer Readings—Sea Level and Altitude	R^2 Ratio Increase Effective Stack Height	$R^{\frac{2}{5}}$ Ratio Increase Stack Diameter
0	30.00	1.000	1.000	1.000
1000	28.88	1.039	1.079	1.015
2000	27.80	1.079	1.164	1.030
3000	26.76	1.121	1.257	1.047
4000	25.76	1.165	1.356	1.063
5000	24.79	1.210	1.464	1.079
6000	23.87	1.257	1.580	1.096
7000	22.97	1.306	1.706	1.113
8000	22.11	1.357	1.841	1.130
9000	21.28	1.410	1.988	1.147
10000	20.49	1.464	2.144	1.165

123 Logically, the draft resistance of an economizer is proportional to the square of the velocity of flue gases, and to the length of economizer. The velocity of gases is directly proportional to the boiler horsepower, with an assumed excess air supply, and inversely proportional to the net area of the economizer, side dampers closed, and likewise proportional to the mean absolute temperature of gases within the economizer. The draft resistance of the economizer for sea level has thus been formulated as follows:

$$D_e = \left(\frac{\text{B.h.p.} \times T}{A}\right)^2 \times \frac{L}{T} \times \frac{J}{10^9} = \left(\frac{\text{B.h.p.}}{A}\right)^2 \frac{T \times L}{10^9} \times J \ldots\ldots [18]$$

in which

D_e = draft resistance of economizer

B.h.p. = boiler horsepower actually developed, a quantity

TABLE 6—PART 1 APPROXIMATE DRAFT RESISTANCE, INCHES OF WATER; STANDARD GREEN FUEL ECONOMIZERS WITHOUT SIDE DAMPERS; 4 SQ. FT. OF ECONOMIZER HEATING SURFACE PER RATED B.H.P. OF BOILER.

Based on 50 per cent Excess of Air Supply; Feedwater Temperature 150 deg., Sea Level, and Other Assumed Data. Oil Fuel.

Draft Resistances in Inches of Water Column for various Widths and Lengths.

Boilers at 100 per cent load of rated capacity.

Temperature gases entering economizer, deg.. 425

Temperature gases leaving economizer, deg.. 275

Average temperature in economizer, deg.. 350

Gases per b.h.p. of boilers, lbs.. 52.8

Assumed b.h.p. of economizer 3 b.h.p. per tube at rating.........b.h.p. $=3\ N-3\ (L\times W)\times 1.00$

Area	16.6		21.85		27.00		32.25		39.25	
Wide	4		6		8		10		12	
L	B.h.p.	*D″*	B.h.p.	*D″*	B.h.p.	*D″*	B.h.p.	*D″*	B.h.p.	*D″*
8	96	0.000282	144	0.000365	....	....	....	....	....	......
12	144	0.000951	216	0.00123	288	0.00144	....	....	....	......
16	192	0.00225	288	0.00292	384	0.00341	480	0.00373	....	......
20	240	0.00440	360	0.00571	480	0.00666	600	0.00729	720	0.00708
24	288	0.00761	432	0.00989	576	0.0115	720	0.0126	864	0.0122
28	336	0.0121	504	0.0157	672	0.0183	840	0.0200	1010	0.0194
32	384	0.0180	576	0.0234	768	0.0273	960	0.0299	1150	0.0290
36	432	0.0257	648	0.0333	864	0.0388	1080	0.0425	1300	0.0413
40	480	0.0352	720	0.0457	960	0.0533	1200	0.0583	1440	0.0567
44	528	0.0469	792	0.0610	1060	0.0709	1320	0.0776	1580	0.0754
48	576	0.0609	864	0.0792	1150	0.0920	1440	0.101	1730	0.0990
52	624	0.0774	936	0.100	1250	0.117	1560	0.128	1870	0.125
56	....	....	1010	0.125	1340	0.146	1680	0.160	2020	0.156
60	....	....	1080	0.155	1440	0.178	1800	0.197	2160	0.191
64	....	....	....	....	....	....	1920	0.239	2300	0.232
68	....	....	....	....	....	....	2040	0.287	2450	0.279
72	....	....	....	....	....	....	2160	0.340	2590	0.331
76	....	....	....	....	....	....	2280	0.400	....	......
80	....	....	....	....	....	....	2400	0.467	....	......

DRAFT RESISTANCE

Correction Factors for Increased Area of Side Dampers

No. of Dampers Open	No. of Tubes Wide				
	4	6	8	10	12
0	1.00	1.00	1.00	1.00	1.00
One Side	0.484	0.564	0.621	0.667	0.769
Both Sides	0.285	0.361	0.423	0.476	0.581

Approximate Correction Factors for 75 Per Cent and 100 Per Cent Excess Air Supply

Per Cent Excess..........	50	75	100
At Rating................	1.00	1.34	1.74
At 1.5 Rating...........	1.00	1.34	1.74

TABLE 6—PART 2 APPROXIMATE DRAFT RESISTANCE, INCHES OF WATER; STANDARD GREEN FUEL ECONOMIZERS WITHOUT SIDE DAMPERS; 4 SQ. FT. OF ECONOMIZER HEATING SURFACE PER RATED B.H.P. OF BOILER.

Based on 50 per cent Excess of Air Supply; Feedwater Temperature 150 deg., Sea Level, and Other Assumed Data. Oil Fuel.

Draft Resistances in Inches of Water Column for various Widths and Lengths.

Boilers at 150 per cent of rated capacity or 50 per cent overload.

Temperature gases entering economizer, deg. 500

Temperature gases leaving economizer, deg. 330

Average temperature in economizer, deg. 415

Gases per b.h.p. of boilers, lb. 54.6

Assumed b.h.p. of economizer 3 b.h.p. tube at rating. . . . . b.h.p. $=3\,N \times 1.50 \times 54.6 \div 52.8$.

16.6		21.85		27.00		32.25		39.25		Area
4		6		8		10		12		Wide
B.h.p.	D″	B.h.p.	D″	B.h.p.	D″	B.h.p.	D″	B.h.p.	D″	L
149	0.000732	223	0.000951	....	....	....	....	....	......	8
223	0.00247	335	0.00321	447	0.00374	....	....	....	......	12
298	0.00586	447	0.00759	596	0.00886	745	0.00972	....	......	16
372	0.0114	558	0.0149	745	0.0173	931	0.0189	1120	0.0184	20
447	0.0198	670	0.0257	893	0.0299	1120	0.0328	1340	0.0318	24
521	0.0314	782	0.0408	1040	0.0476	1300	0.0520	1560	0.0505	28
596	0.0468	893	0.0609	1190	0.0708	1490	0.0776	1790	0.0754	32
670	0.0667	1010	0.0868	1340	0.101	1680	0.111	2010	0.107	36
745	0.0915	1120	0.119	1490	0.138	1860	0.152	2230	0.147	40
819	0.122	1230	0.158	1640	0.184	2050	0.202	2460	0.196	44
893	0.158	1340	0.205	1790	0.239	2230	0.262	2680	0.255	48
969	0.201	1450	0.261	1940	0.304	2420	0.333	2903	0.322	52
....	....	1560	0.326	2090	0.379	2610	0.416	3130	0.406	56
....	....	1680	0.401	2230	0.465	2790	0.512	3350	0.497	60
....	....	....	....	....	....	2980	0.619	3570	0.601	64
....	....	....	....	....	....	3160	0.745	3800	0.724	68
....	....	....	....	....	....	3350	0.884	4020	0.859	72
....	....	....	....	....	....	3540	1.037	....	......	76
....	....	....	....	....	....	3720	1.212	....	......	80

Table 6 has been computed by the use of the following empirical formula:

$$D = \frac{P_0}{P_1}\left(\frac{\text{B.h.p.}}{A}\right)^2 \frac{T \times L}{10^9} \times J$$

in which

D = the loss in pressure through the economizer expressed in inches of water column
P_0 = normal barometer, sea level
P_1 = barometer at the altitude of plant
B.h.p. = B.h.p. rating of standard economizer as assumed above
A = net area in sq. ft. between tubes for the flow of gases
T = average absolute temperature in deg. fahr. of the gases in the economizer
L = the length of economizer in rows of tubes
J = a coefficient determined from tests

For the purposes of this table the value of J is taken as 1.3, an average of values from a number of tests ranging from about 50 per cent above to 50 per cent below this figure. For 75 per cent and 100 per cent excess air supply, see approximate correction factors. If the volume of excess air due to leakage is sufficient to change the temperatures assumed, the correction factor for excess air will be correspondingly changed. B.h.p. refers to work done by boilers only, based on feedwater temperature entering the boilers, and not that entering economizers.

Draft resistance is based on areas of standard economizers with standard spacing and tube length. No allowance has been made for the resistance of any additional breeching or turns made necessary by the use of the economizer. Such resistances must be added to the values given in table.

For altitude D should be increased inversely with the barometric reading.

See footnotes, Table 7, regarding air supply greater than 50 per cent excess, leakage through brick settings, etc.

proportional to the weight of gases flowing per minute

A = net area through economizer, side dampers closed

T = mean absolute temperature, deg. fahr., of gases within economizer

L = number of sections of tubes in economizer

J = a constant, assumed as 1.3

124 The quantity D is, in reality, a draft coefficient based on the fundamental quantities and, neglecting J, should be modified by a correcting factor, depending on the designer's data. Based on the writer's data, this coefficient averages about 1.3 and varies from 0.65 to 2.0; J is therefore made 1.3.

125 Table 6 gives approximate resistances for various economizers, based on boiler horsepower ratings corresponding to 100 and 150 per cent load on boilers and economizers, and indicates the assumed temperatures entering and leaving the economizer, ratio excess air, etc. It is important to note that the boiler horsepower indicated represents the actual boiler horsepower developed by the boiler and not the rated boiler horsepower, a quantity for purposes of estimating the resistance of the economizer strictly proportional to the actual weight of gases flowing per second.

126 The fundamental equation [6] of draft resistance is modified, due to installation of economizers, by the introduction of an added term covering the resistance of economizers and a correction of chimney temperature. The equation [12] for boiler horsepower for Case III, central chimneys, is then written

$$\text{B.h.p.} = \frac{60^2}{G} \sqrt{d^5} \sqrt{\frac{fKH - (D_1 + D_2 + D_e)}{2.3\,T\,(H + 1.5d)}} \ldots\ldots\ldots\ldots [19]$$

where K is dependent on economizer outlet temperature.

127 Equation [19] is rendered approximate, as the term for breeching resistance is now made dependent on economizer outlet temperature; however, the breeching in a well designed economizer plant should be less than 10 diameters in length and these two errors may compensate. If added flue turns are introduced, then, for close figuring, [19] must be corrected accordingly.

128 Owing to the special nature of economizer installations, variation in size and arrangement of economizers, etc., it is not practicable to compute a general table of chimney capacities for economizer plants. If a chimney of minimum first cost be selected

for the given boiler conditions, ignoring the installation of economizers, then, to provide for economizers it should merely be necessary to increase the height and not the diameter of chimney to compensate for the added resistance, and lowering of stack temperature. This increase in height may be determined approximately, based upon the known increase in draft resistance and reduction of chimney temperature, from the draft tables. It is not possible to compute any simple table setting forth accurately the ratios of increased chimney heights for the various stated conditions, based on chimney capacities without economizers, by reason of the involved breeching resistances as well as stack resistances and varying ratios of total chimney draft required to overcome chimney friction.

129 It is a simple matter to compute for central chimneys a table of ratios giving the reduction of total boiler horsepower due to the installation of economizers, which, in connection with Table 4, enables a ready determination of chimney sizes. Table 7 has been computed on this basis and has reference to the capacities given in Table 4 for central chimneys without economizers.

130 It is self-evident that the minimum heights given in Table 4 will not furnish sufficient draft for any reasonable capacity, with the added friction and cooling of gases due to economizers. Table 7 indicates the minimum heights consistent with reasonable chimney proportions.

131 In order to obtain a chimney of minimum first cost when an economizer is used, trial and comparison should be made of several sizes.

GENERAL CONSIDERATIONS

132 The chimney capacities set forth in tables are conservative for the exact conditions stipulated, although the margin for safety is not so great as might ordinarily be presumed from the approximate nature of chimney calculations.

133 These tables will not have application to all types of boilers and all conditions of operation and refer specifically to boilers of the highest type and under favorable conditions of operation. If the designer is uncertain as to the conformity of his conditions to those stipulated in tables, due margins for safety should be allowed for selecting chimney sizes.

134 While the writer has obtained results of operation for

TABLE 7 SHOWING APPROXIMATE RATIOS OF CAPACITY OF CHIMNEYS WITH ECONOMIZERS TO CAPACITY OF SAME CHIMNEY WITHOUT ECONOMIZER BASED ON ASSUMPTIONS BELOW

	At Rated Load									150 Per cent of Rated Load								
	Initial temperature of gases in stack without economizer, deg.								425	Initial temperature of gases in stack without economizer, deg.								500
	Initial temperature of gases in stack with economizer, deg.								275	Initial temperature of gases in stack with economizer, deg.								330
	Db. Draft at chimney side of boiler damper, in.								0.15	*Db.* Draft at chimney side of boiler damper, in.								0.30
De	0″	0.005″	0.010″	0.025″	0.05″	0.10″	0.15″	0.20″	0.30″	0″	0.005″	0.010″	0.025″	0.05″	0.10″	0.15″	0.20″	0.30″
H	Ratio of Chimney Capacities									Ratio of Chimney Capacities								
60	0.502	0.439	0.365															
70	0.669	0.638	0.608	0.500	0.233													
80	0.737	0.717	0.696	0.630	0.500													
90	0.774	0.759	0.743	0.694	0.605	0.364				0.495	0.455	0.411	0.233					
100	0.798	0.786	0.774	0.735	0.667	0.498	0.233			0.645	0.623	0.600	0.526	0.371				
110	0.815	0.805	0.794	0.762	0.705	0.575	0.406			0.715	0.699	0.684	0.634	0.542	0.275			
120	0.827	0.819	0.810	0.782	0.734	0.628	0.497	0.321		0.759	0.748	0.736	0.698	0.630	0.465	0.189		
130	0.837	0.830	0.822	0.798	0.756	0.665	0.560	0.428		0.788	0.778	0.769	0.738	0.684	0.561	0.401	0.080	
140	0.844	0.838	0.830	0.809	0.772	0.693	0.603	0.498	0.126	0.809	0.801	0.795	0.767	0.721	0.623	0.504	0.347	
150	0.851	0.844	0.838	0.818	0.785	0.715	0.637	0.548	0.299	0.825	0.818	0.811	0.789	0.750	0.666	0.571	0.456	
160	0.855	0.850	0.844	0.826	0.797	0.733	0.664	0.586	0.388	0.837	0.831	0.825	0.805	0.771	0.699	0.619	0.526	0.254

The capacity ratios given above are applicable to chimney capacities given in Table 4 for boiler plants without economizers, chimneys centrally located sea level, and are subject to all of the limitations stated for Table 4. *De* =draft resistance due to economizer, for values of which see Table 6. If economizer layout introduces greater breeching resistance than assumed in development of Table 4, *De* should be increased accordingly to cover additional length of and turns in breeching. Table 4 is based on straight run of breeching. B.h.p. refers to work done by boilers only, based on feedwater, temperature entering boilers, and not that entering economizers. Boiler efficiencies are as stated in Table 4, not including economizer saving. Ratios given for various heights enable the maintenance of same net draft at boiler outlet for stated boiler ratings, making necessary allowance for loss of draft due to cooling of chimney gases and interposed economizer resistance. For sake of uniformity with other tables, ratios are based on 50 per cent excess air supply at maximum load measured in chimney. This condition is realizable in small plants having masonry settings and in large plants with steel breechings and steel economizer casings. For large plants, with brick flues and brick economizer settings, air leakage may increase relative volume of chimney gases from 20–40 per cent or more. *De* should be modified accordingly for resistance, and the then calculated B.h.p. of chimney reduced to allow or air leakage. Correction must also be made for modified chimney temperature.

numerous chimneys burning oil fuel, it is generally the rule that oil-burning chimneys in actual service are underloaded and that comparatively few chimneys are operating at their maximum capacity. Therefore, in computing chimney capacities given it has been the writer's aim to determine from test observations the actual boiler resistances for the various conditions of operation, and from this starting point to calculate chimney height sufficient to produce the required draft at boiler, and diameter sufficient to carry away the necessary volume of gases, with the margin of draft available for the friction of flow through breeching and chimney.

135 Where building conditions demand chimneys of considerable height the draft required at the boiler outlet becomes a smaller proportion of the potential draft of the chimney, and accordingly an error in the assumption of the draft resistance of the boiler will have less effect on the total capacity of the chimney of given dimensions.

136 When selecting chimneys of reduced height for boilers operating at or near rating the draft resistance of the boiler for least cost and economical operation requires a larger proportion of the potential draft of the chimney. An error in the assumption of draft resistance of the boiler largely affects the capacity possible with a chimney of given dimensions.

137 In all instances the diameter of the chimney and frictional loss within the chimney and breeching have been based on 50 per cent excess air over chemical requirements. For a number of years the writer thought it necessary to provide for not less than 100 per cent excess air; but from actual experience and data from a number of tests where the firing conditions were directed by ordinary operators, it has been shown that 50 per cent excess air is sufficient for good practice under maximum load conditions in first class installations.

138 The various features affecting the draft resistance of a boiler have been pointed out, many of these being subject to variation by the local operator. The greatest influence on draft resistance, however, is that due to excess air. Under ideal conditions it is possible to fire boilers with approximately 10 per cent excess air and even less, giving a corresponding reduction in the draft resistance of the boiler. As the excess of air supply is reduced, the capacity of a chimney increases in a multiplied ratio, due, not only to the decreased volume of gases to be handled and

the decreased weight of gases per b.h.p. by reason of greater boiler efficiency, but also to decreased draft resistance of the boiler and the greater draft available for the flow of gases through breeching and chimney. It will therefore be possible not only to point to chimneys of the dimensions given in tables showing a greater capacity than stated corresponding to reduced air supply, but also of less capacity than stated corresponding to increased air supply.

139 In considering boilers of differing types, the chimney dimensions will depend largely on the boiler design and efficiency. particularly with respect to the draft resistance of the boiler and the minimum air supply as dependent on furnace design. Certain types of boilers will show less draft resistance and others more than indicated for Babcock and Wilcox boilers, with a corresponding variation in chimney capacity.

140 Many inexperienced in the burning of crude oil carry the impression that the mere closing of the damper will result in the reduction of excess air supply practically to the theoretical limit. It is actually possible to cause the formation of smoke with considerably more than 50 per cent excess air supply in a poorly designed boiler furnace. The furnace must have a particular shape, volume, and arrangement of checkerwork with respect to the burner to permit the complete admixture of air for combustion and vaporized oil; and the furnace efficiency with respect to this admixture, influenced by the surrounding heated brickwork, determines the minimum excess air permissible without undue smoking.

141 In the calculation of chimney tables it was deemed best not to provide sufficient margin for safety in order to cover all possible conditions of operation, but rather to establish as a benchmark the maximum chimney capacities based upon the draft resistances obtained during conditions of uniform load, and to modify capacities as need be for the special conditions of operation, which should be carefully studied before selecting chimneys for a given installation.

142 When operating with 50 per cent excess air, uniform load, and furnaces properly adjusted, there should be a slight haze from each boiler; on uniform load there should not be an objectionable amount of smoke from central chimneys connected to a number of boilers, except when fires are unequally adjusted.

As is well known, automatic control of fires insures an equal division of load among all boilers in service.

143 It is possible for short intervals to force the chimneys beyond their maximum capacity, giving rise to increased smoke density; but good practice does not sanction such heavy smoking, except for the inevitable short intervals on swinging loads.

144 In passing, it may be stated that it is common practice to gage the air supply roughly by the density of smoke emitted from the chimney, and it is wise in boiler plant design to provide openings through the roof so that the firemen can conveniently observe the chimneys.

145 Tables provide for the slight reserve necessary to regulate fluctuating steam pressure with careful firing, but do not provide sufficient reserve to restore steam pressure rapidly after a material pressure drop. During such intervals careless firing may require a very large momentary overload on boilers. While a certain draft margin is necessary, a chimney which makes necessary careful attention of furnaces is a positive benefit from the standpoint of boiler abuse and cost of tube renewals.

146 If a number of boilers be connected to a central chimney subject to hand-firing, certain boilers under unequal adjustment of fires will be operating at overload and others at underload. To offset these conditions and to maintain the maximum total capacity required, it is necessary slightly to increase the height of stack, and in the table no margin is specially provided to cover this contingency. When firing under automatic control, which insures an equal division of load among all boilers in service, such margin for increase is unnecessary.

147 It has been found in large plants having central chimneys and subject to heavy swinging loads, that on a sudden increase of load there is a sluggishness of chimney action due to inertia of gases within the chimney and breeching, as well as a time element required for the heating of the chimney; smoke is usually formed during such intervals. Particularly where flues are of masonry construction it is desirable to increase the height of chimney slightly to counteract this influence, although occasional smoking cannot even then be completely eliminated.

148 For swinging loads the writer would reduce the tabulated capacities for central chimneys from 10 per cent to 20 per cent, this factor being one of judgment and not based on experimental data.

149 When boilers have been operating for a period at reduced load, on a sudden increase in load owing to the cooling action of the furnace, an increased air supply will be required over uniform load conditions in order to obtain the desired capacity without smoking. This condition will continue until the furnace brickwork has been thoroughly heated, and common practice permits smoking during this short interval.

150 Consideration should be given to a certain reserved draft in order to insure, at all loads, a positive draft within the setting by throttling the air admission. It has been pointed out by Dr. Jacobus that under certain load conditions, should a negative draft within the setting occur, there will be a tendency for the hot gases to pass outward through the setting making possible a considerable overheating of the brick furnace. On the contrary, with a positive draft within the setting, a certain amount of air infiltration will tend to cool the brickwork and thus increase the life of the setting. In general, the tendency to form a negative draft diminishes with overload.

151 In the design of chimneys it is necessary to give careful consideration to the question of minimum air supply by reason of the tendency of increased smoking with changes of load when operating near the theoretical limit. Sooting of boiler tubes might easily reduce the heat transfer of the boiler to such an extent as to more than offset the economy due to decreased air for combustion.

152 A well designed central chimney subject to variation of number of boilers in service requires, for maximum economy, a comparatively small amount of stack and breeching friction in order to insure what might be termed "draft regulation." In boilers subject to hand adjustment or, for that matter, automatic control of air supply, it is desirable to have a nearly fixed draft at the boiler outlet for each addition of load, with the least possible change due to variation in the number of boilers in service. It is self-evident that this variation in draft will give a variable air supply, with a fixed position of damper for each load, and thus reduce the fuel economy of the plant.

153 The proper determination of chimney height, with respect to the maximum draft required at boilers, will have a marked influence on the fuel economy of the plant. While a certain draft margin is necessary for the special conditions of plant operation, it should at all times be the aim to provide a chimney

with the least possible excess draft over the necessary maximum.

154 In plants subject to uniform factory or pumping loads approximately at rating, the chimney height should be very much reduced and such that when boilers are operated with nearly wide open dampers the air supply will not materially exceed the requirements for economical firing.

155 With excessive height of chimney it is not possible to open the damper more than a fractional amount, thus giving a large increased air supply for a very small added opening of damper, making economical operation at variable loads impossible with hand control, and more difficult with automatic control.

156 The influence of excessive draft becomes more apparent when boilers are operated for intervals at reduced capacity. If the firemen be careless and leave the dampers and ash doors unchanged during variations of load, it is apparent that the draft resistance of the boiler becomes nearly a constant, and except for a slight decrease in the temperature at reduced loads, the tendency is toward a practically constant flow of gases through the setting regardless of the rate of firing, and correspondingly, an increasing ratio of excess air at light loads. While with tall chimneys the excess air supply is abnormal even at maximum loads, it becomes thus increasingly wasteful on reduced loads.

157 In one instance which came to the writer's observation, a chimney gave 300 per cent excess air when operating at rating, and 450 per cent excess air when operating 30 per cent load, the dampers and ash doors being wide open in both instances. In the latter instance the resulting boiler efficiency was less than 40 per cent, whereas, even at this reduced load, the efficiency should have been approximately 75 per cent.

158 Experience has indicated that plants having tall chimneys subject to hand control are, in general, uneconomically operated; while plants having relatively low chimneys indicate a superior performance, due largely to the decreased margins for excess draft and possible wastage of fuel with excess air supply. This fact is particularly impressive after an examination of the economy attainable in power plants in connection with tall office buildings, where high chimneys are a necessity. In such instances a reduction of stack diameter to absorb by stack and breeching friction a considerable portion of the excess draft will result in an improved economy of plant.

CHIMNEYS FOR AUXILIARY STEAM POWER STATIONS

159 It has been pointed out that chimneys of reduced height are possible under certain conditions of operation, namely, with uniform load, approximately at rating, but the impression should not prevail that tall chimneys are unnecessary for oil firing under certain conditions of heavy load.

160 In auxiliary steam power stations it will be found necessary at times to force boilers to approximately 100 per cent overload. Under these conditions, due to the rapid increase of draft resistance of boiler, the heights of central chimneys will vary, depending on the type and size of boiler, plant arrangement, etc., from less than 100 ft. to about 150 ft. for sea-level conditions, and even greater heights will be required at altitudes.

161 The height may be somewhat reduced below that otherwise necessary on heavy overloads by increasing judiciously the admission area through floor of furnace. On large boilers reduction of draft and chimney height at overloads may also be effected by the provision of larger air admission doors, limited, however, by the creation of a negative draft within setting, as already discussed.

162 It is important to note that the chimney heights stated for certain overload capacities apply only when a sufficient number of the connected boilers are in operation to maintain the stated initial chimney temperature, and to prevent undue cooling within the chimney. In auxiliary steam power plants, or other stations where at times comparatively few of the boilers may be in operation on a central chimney, this feature must be considered.

163 If the temperature of the water is, say, 100 deg. fahr., or less, within a boiler connected to a cold chimney, then, on an emergency call, a considerable period will elapse before the gases entering the chimney will reach such temperature as to produce an appreciable draft and make possible rapid firing. If, as is ordinarily the case, all the boilers be kept under steam by intermittent and alternate firing, the temperature of the boiler heating surface will be considerably above the boiling point, and the mere opening of the boiler dampers will cause a flow of air through the setting, the incoming air being heated by the hot water within the boiler tubes. The starting of the fire will grad-

ually increase the temperature of the escaping gases, but owing to the reduced draft, a considerable interval of time will elapse before the boiler will be operating at such rate of firing as to give the necessary increase in temperature of escaping gases and chimney draft for any appreciable overload on the boiler. Under these conditions it is evident that the taller the chimney the less the required time for getting under way; also, that an independent chimney will, in general, enable quicker starting than a central chimney when firing a fractional number of boilers. On the other hand, if a central chimney be employed, and one or more boilers are constantly under fire, the chimney will at all times have a relatively higher minimum temperature, and by reason of the draft produced, enable the quicker starting up of a large number of connected boilers than is possible with independent chimneys.

164 As service demands become more exacting it is probable that auxiliary steam power stations will ultimately be provided with forced draft fans to supply air for the rapid starting of boilers which are to be used for the first few minutes, and until boilers, furnaces, flues and chimneys are thoroughly heated, thus following, to a certain extent, battleship practice; or steam jet chimney blowers may be used while starting.

EXAMPLES FROM PRACTICE

165 *Plant No. 1.* A plant of three Babcock and Wilcox boilers, 7 high, total 275 h.p.; one Green fuel economizer 6 by 20; one side damper; one pair of induced draft fans; one 42-in. discharge stack; 54-ft. height above floor line. Plant initially installed for coal burning. When fired with oil fuel, at rating and at slight overloads it was found that the natural draft from chimney was sufficient to produce the necessary draft at boilers to overcome resistance of economizers and resistance of blades of fan wheels with fans at rest. When a wind storm reduced the height to 40 ft. the capacity was still ample. A second wind storm reduced the height to 19 ft. which was ample at light loads, but required assistance of fans at 180 r.p.m. at rating.

166 *Plant No. 2.* A plant having a number of 9 high Babcock and Wilcox boilers; after the San Francisco earthquake of 1906, was able to operate at rating with chimney height reduced by earthquake to 25 ft. above floor line.

167 *Plant No. 3.* A plant of 12 high Babcock and Wilcox boilers, with horizontal short breeching and steel chimney 115 ft. above floor line; sea level; chimney diameter relatively large and friction small; three boilers connected to one chimney, one boiler, on test, operated at 210 per cent of rated capacity, test conditions.

168 *Plant No. 4.* A plant of 12 high Babcock and Wilcox boilers; a total of 15 boilers, connected to a central chimney 125 ft. above floor line; Green fuel economizers; induced draft fans; one boiler operated at 208 per cent of rating, other boilers at about rating. The ordinary draft of chimney was insufficient, and necessitated the use of fans. Economizers 80 tubes long, resistance excessive.

169 *Plant No. 5.* A plant of 12 high Babcock and Wilcox boilers, 8 boilers connected to a central chimney; Green fuel economizers; panel system. Chimney reduced by earthquake to 70 ft. Natural draft sufficient for operation of boilers at about rating, but insufficient for overload.

170 *Plant No. 6.* A plant of 14 high Babcock and Wilcox boilers; central chimney; large diameter; approximately 70 ft. height above floor line. Maximum capacity one boiler under test, when other boilers operating at about rating, natural draft 145 per cent of rating.

171 *Plant No. 7.* Stirling boilers, 250 h.p., connected to independent overhead chimney 80 ft. height above floor line. Maximum recorded capacity, but not maximum possible capacity, 90 per cent overload, still showing an excess air supply; plant operating factory load, boilers intended for operation at rating only; on account strong draft from chimney could not get fireman to operate boilers at rating. Finally reduced chimney height to 40 ft. to prevent such excess air through furnaces as would prevent excessive overloads on boilers. In this manner capacity considerably reduced, but still able to obtain greater capacity than rating. Exact overload not determined. Plant economy increased by lowering the temperature of escaping gases and amount of excess air.

172 *Plant No. 8.* Battery of two Stirling boilers 250 h.p. each, connected to a central overhead chimney with A-shaped breeching, the total height being 40 ft. above floor line. Intended for use in oil fields with unskilled firemen. Height fixed at 40 ft. to prevent overload on boilers and the waste of fuel from excessive overload and from excess air. Boilers have operated at capacities in excess of rating.

173 *Plant No. 9.* Office building plant. Stirling boilers, chimney height approximately 200 ft. on account of excess draft even with partially open dampers and air doors excess air over 200 per cent at rating, probably 15 to 20 per cent waste of fuel due to excess air.

THE INTERNATIONAL SMOKE ABATEMENT EXHIBITION

HELD IN LONDON, MARCH AND APRIL 1912

By George H. Perkins

ABSTRACT OF PAPER

The solution of the smoke problem must necessarily rest with the mechanical engineer. The development of a strong and intelligent public sentiment on the economic and sanitary necessity for smoke abatement, however, must come through the widespread campaign of education in which the Society should take an active part. Societies for this specific purpose are doing much effective work in England with most excellent results. The lines along which these organizations are working are briefly indicated in this paper, which is an informal report made originally to the trustees of the Lowell Textile School, Lowell, Mass., who delegated the writer to attend the International Smoke Abatement Exhibition and Conference held in London from March 23 to April 4, 1912.

THE INTERNATIONAL SMOKE ABATEMENT EXHIBITION

HELD IN LONDON, MARCH AND APRIL 1912

By George H. Perkins, Lowell, Mass.

Associate Member of the Society

An International Smoke Abatement Exhibition and Conference, under the auspices of the London Coal Smoke Abatement Society, was held at the Royal Agricultural Hall, Islington, London, March 23 to April 4, 1912.

2 The object of the exhibition was to make a comprehensive display of the most approved modern methods and devices for the abatement of coal smoke, applicable either to industrial plants or domestic fires. There were also exhibits showing the injurious effects of smoke deposits upon health, building materials, works of art and vegetable life, with the view of creating an active and intelligent public sentiment on the importance of this widespread nuisance.

3 The educational work was supplemented by a series of conferences which were well attended by official delegates from most of the principal cities of England and Scotland as well as from Germany, Holland, Sweden, and other foreign countries.

4 The exhibition attracted much favorable attention from the press and general public, and upon the whole was considered a success. The great coal strike which was at its height at the time, while interfering with the attendance and delivery of exhibits to some extent, tended to increase public interest in the affair and to divert attention to the larger question of the conservation of the coal supply.

PRESENT STATUS OF THE SMOKE PROBLEM IN ENGLAND

5 Smoke has been a recognized nuisance in England for nearly a century. As early as the year 1819, Parliament ap-

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The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

pointed a committee to investigate this question which reported that effective smoke abatement devices existed and should be used. The public health acts of 1875 and 1891 included sections covering smoke abatement which are the only present laws on the subject. These statutes have not proved satisfactory as they are not sufficiently definite and do not make provision for their proper enforcement.

6 Practically all of the effective work done in abating the smoke nuisance has been accomplished during the past 15 years. The great improvement in atmospheric conditions, as far as London is concerned, has been due principally to the following agencies:

a The electrification of all of the underground railways formerly operated by coal burning locomotives.

b The rapid increase in the use of gas appliances for heating and cooking. When authorities agree that more than one half of London's smoke comes from domestic fires burning soft coal, the following figures are significant:

Year	Number of gas appliances in use
1891	46,000
1896	223,000
1901	445,000
1906	989,000
1911	1,494,000

c The work of the London Coal Smoke Abatement Society, a voluntary association which has for its objects the following:

1 To aid in enforcing the existing law through the local sanitary authorities, by the employment of inspectors who, after investigation, shall report offenders to the proper officials.

2 To promote and encourage all voluntary efforts to abolish smoke from private dwellings and to investigate the best means for effecting this object.

3 To obtain particulars and evidence of methods of dealing with smoke nuisances at home and abroad.

4 To publish information on matters relating to smoke nuisances, stimulate invention through the offer

of prizes and to conduct tests on heating, cooking and stoking apparatus.

5 To effect the amendment of the present laws with the object of making them more efficient.

7 Other societies of the same character have been organized in some of the larger cities and are doing effective work along similar lines to that of the London society. One of the most active of these is the Smoke Abatement League of Great Britain which has its headquarters in Manchester and branches in many of the other large industrial centers.

8 The most direct evidence of the improvement made in smoke abatement in recent years is in the record of observations of atmospheric conditions taken in many of the large cities. The "black fogs" once so prevalent in London and which have been proven to have been due largely to smoke are to-day practically unknown. The winter sunshine of London is to-day about 40 per cent of that observed in the country districts, which is a figure double that of 30 years ago.

9 In nearly all of the larger cities a marked improvement along similar lines has been made each year and through the agencies already mentioned. In the manufacturing districts particularly, the boiler users are taking increasing interest in this matter, realizing that it has an important influence upon the efficiency of their plants. The public is also aroused to the situation and the urgency and practicability of smoke abatement seem to be generally appreciated.

SUMMARY OF PAPERS PRESENTED AT THE CONFERENCE

10 The papers presented at the conference may be grouped under the following heads:

A Smoke Pollution
    1 Economic and Artistic Aspects
    2 Effects on Plant Life
B Smoke Abatement
    1 Organization of Preventive Action
    2 Physical Principles of Smoke Abatement
C Laws and Legislation
    1 Proposed New Legislation
    2 Administration of Existing Law

(*A*-1) *Economic and Artistic Aspects*

11 Under this head papers were read treating of the effects of smoke upon building materials, mural and house painting, ornamental and structural metal work and also the measurement of soot deposits.

12 Examples were cited of the disintegration of the stone in various buildings due to sulphuric acid in the atmosphere. This acid is formed from the union of the oxidized sulphur compounds, resulting from the combustion of coal, with the moisture of the atmosphere. An analysis of stone taken from St. Paul's Cathedral showed 74 per cent of calcium sulphate, the original stone being calcium carbonate or limestone. Similar results were obtained on samples taken from Westminster and other well known and historic structures. Limestone appears to suffer more than any other variety of stone while granite is not seriously affected.

13 The effect of smoke on mural and fresco paintings was due to a similar action, the plaster beneath the pigments losing much of its binding power by the chemical change taking place. The effect upon house paint is also very pronounced, particularly on the lead paints. Practically all outside work must be repainted at least once a year. Zinc oxide paints were shown to be more stable than lead pigments against the action of atmospheric acids.

14 The most serious effect upon metal work is the corrosion of structural steel. Several collapses of iron roof trusses, notably that of the Charing Cross Station, have resulted from this cause. In some instances it was found that about 10 per cent of the iron had been dissolved into ferrous sulphate. The corrosion of uncovered iron or steel wire has also been observed and found to be extensive.

15 During the past year the soot fall of London has been carefully measured by means of specially devised soot gages. This work has been in charge of the officers of the Smoke Abatement Society and the extraordinary results obtained may be summarized as follows:

16 The total yearly deposit from the atmosphere was 650 tons per square mile, or a total of 76,050 tons per annum for the entire administrative county of London of 117 square miles. This figure includes 8000 tons of sulphates, 6000 tons of ammonia and 3000 tons of chlorides, the balance being carbon and tarry prod-

ucts. The deposit per square mile at Surrey, on the border of the metropolitan area, was only 195 tons per year, or less than one third that of London proper, showing clearly the comparative purity of country air.

(*A*-2) *Effects on Plant Life*

17 Evidence was given by the authorities in charge of the Kew Gardens and other parks as to the difficulty of maintaining vegetation in smoke affected districts. Only a few hardy plants and shrubs will survive the winter season in the large cities. The main effects of smoke on vegetation are due to the following:

1 Reduction of light and heat from the sun.
2 Soot deposits excluding still more light.
3 Tarry deposits blocking the pores of the plants.
4 Acid deposits lowering the vitality of plants.

18 The effect upon health of a smoke polluted atmosphere was clearly shown by figures given from Glasgow, where a study has been made of the effect of smoke fogs upon the death rate from bronchial diseases such as bronchitis, pneumonia and pleurisy. The deaths from these diseases alone increased at a remarkably rapid rate during prevalent fogs and could be traced to no other source.

(*B*-1) *Organization of Smoke Abatement Work*

19 The work of the various smoke abatement societies was described in detail and many suggestions were made regarding the possible future study of this problem. The majority of these suggestions were along the line of determining the actual economic loss to the community caused by smoke, it being generally conceded that the engineering side of the question is well in hand and sufficiently well developed to meet practically all conditions provided careful study is made of the factors involved.

20 Some of the elements in the economic loss to a city are as follows:

1 Added cost of artificial illumination.
2 Added cost of painting exteriors and interiors.
3 Added cost of laundering and other cleansing.
4 Damage to goods in stores and factories.
5 Effect of gloom upon the efficiency of workers.

21 Recognition was made of the splendid work done on smoke abatement in Cleveland, Ohio, and Chicago, Ill., where the progress made within the past five years has been remarkable. In

both of these cities the problem is handled by efficient departments of the municipal governments furnishing a most excellent example of honest, effective and thorough public service. These two cities are without question well in advance of any other American municipalities in the matter of smoke abatement.

(*B*-2) *The Physical Principles of Smoke Abatement*

22 The papers presented under this head covered various phases of the subject, including the effect of smoke upon steel making, losses due to incomplete combustion, hand and mechanical firing, the manufacture of smokeless fuels and the construction and operation of producer gas plants. The production of steel without objectionable smoke was held to be practicable, without affecting the quality of the steel.

23 Results of tests were given showing the loss due to unburned combustible matter in smoke and that due to a non-conducting layer of soot upon the surfaces of boiler tubes and plates.

24 The value of trained firemen was especially emphasized and the subject of hand firing and conditions under which it might be successful were discussed. It was proposed to add to the present license for firemen, a certificate of competency based upon a knowledge of proper conditions for perfect combustion, heating value of fuels, etc.

(*C*) *Laws and Legislation*

25 The larger part of the matter presented under this head would not be applicable in this country for obvious reasons. A strong plea was made, however, for uniform and definite legislation on smoke abatement. The principal paper in this group urged the appointment of a royal commission to inquire generally into the subject of smoke emission for the purpose, if possible, of having a stringent general law passed by Parliament. It was suggested that this royal commission, if appointed, be directed to inquire:

1 What standard, if any, is desirable as to the color or density of the smoke which should be deemed a nuisance?

2 How the color or density of smoke can best be identified?

3 Are the fines at present inflicted upon offenders sufficiently adequate to act as deterrents?

4 Who should be punished, the owner of the works, the

engineer in charge, or the stoker, when smoke nuisances result from the careless manipulation of well equipped installations?

5 Should any industries receive special treatment?

6 Should Government inspectors be appointed to act in regard to the emission?

7 Should Government premises be placed under the obligation to prevent smoke nuisances?

8 Ought any hourly or other time limit be fixed during which black or other smoke may be permitted to issue from factory chimneys?

9 Should all cases be heard by stipendiaries? (Paid magistrates.)

10 Can the issue of an unreasonable quantity of smoke be prevented except in cases of accident or breakdown?

11 Can it be shown that the installation of proper appliances generally results in economy as is contended by "smoke abaters"?

12 The incidence, intensity and duration of fogs and their effects on health.

13 The influence of smoke on health.

14 The damage of smoke to buildings, works of art and property generally.

15 Foreign efforts to cope with the smoke nuisance.

16 How far is it possible by smoke abatement to conserve the coal supply?

17 How far is it practicable to deal with the smoke from private dwelling houses?

CLASSIFICATION OF EXHIBITS

26 The exhibits may be classified into the following groups:

a Boiler furnaces; special grate bars and furnace construction; mechanical stokers of all types; smoke preventing devices of various kinds; fuel economizers; briquette making machines.

b Suction gas plants; gas and oil engines.

c Open coal-fire grates; stoves and ranges of all kinds; draft regulators and other smoke preventing appliances for domestic fires.

d Gas stoves and grates; gas heating and cooking appliances.

e Electric heating and cooking appliances.

f Testing devices, including anemometers, tintometers, soot gages, automatic gas analyzing devices, etc.

g Fans and other appliances for the removal of dust of various kinds from industrial plants.

h Smokeless fuels, both natural and artificial.

27 While there was little essentially new in the principles underlying the various devices exhibited, there has been a marked improvement made within a few years in their design and mechanical construction. There was also evidence of an appreciation and careful study of the theoretical side of the question which has undoubtedly contributed largely to the relatively better results obtained in recent years. The exhibits were so numerous and varied that only the most important will be mentioned.

MECHANICAL STOKERS

28 The stoker which appeared to be considered the best adapted for the prevailing conditions in England is the chain grate type.

29 When used with the proper fuel, preferably bituminous slack, this type appears to meet the requirements of smokeless combustion more nearly than any other device, particularly with water tube boilers. The best sample of this stoker exhibited was by the Babcock and Wilcox Company.

30 A number of types of coking stoker were shown, the best being those made by J. Hodgkinson of Salford and E. Bennis of Bolton. These stokers are better adapted to the Lancashire type of boiler, since they are not readily forced. In their action a reciprocating ram, mechanically driven, and located in the bottom of the hopper, forces a small quantity of fuel on to a " dead " or coking plate at each forward stroke. While the fuel remains upon this plate the volatile products are driven off and the coked coal then drops upon the grate bars which are of a sliding type and actuated by a cam motion on the front end. The mass of the fire is constantly carried toward the rear of the furnace by an intermittent and simultaneous movement of all the bars, the return stroke of each bar being independent.

31 There were a number of forms of sprinkler stoker shown

which, although of ingenious design, have not proved satisfactory where smokeless operation is desired.

32 While there were no underfeed stokers exhibited, there are a number of English machines of this class giving satisfaction under the proper conditions. The best known are the Vickers, Bennis, Underfeed and the Proctor. This type of stoker is not so well adapted for use with the average fuel available in England and is not so extensively used as in America.

PATENT FURNACES

33 There were many exhibits under this head, only a few of which possessed real merit. The majority of these appliances consisted simply of properly controlled steam jets with special forms of grate bars. There were a few good devices for the automatic regulation and timing of the supplementary air supply, both over the fire and below the grate. The best of these was the Kowitzke furnace which was excellently designed and simple in operation.

34 Various types of furnace construction, designed to effect a better mixing of the gases, were also shown. The Gregory patent furnace makes use of a supplementary jet of vaporized fuel oil ejected over the fire, which is claimed to add materially to the efficiency.

35 A special boiler known as the Bettington is designed to allow the use of a mixture of air and pulverized fuel or coal dust, introduced by air blast at the bottom of the boiler which is of the vertical type. This boiler is intended for use at mines or wherever fuel of a finely divided character is available.

MISCELLANEOUS DEVICES

36 A device for the optical analysis of the gases of combustion by means of an interferometer was shown by Carl Zeiss and Company. It is based upon the principle that the refractive power of the gases is dependent upon the content of carbon dioxide. A gas refractometer is so calibrated that the analysis of the gas may be read on the scale by simple inspection. A special form of tintometer was also shown in a portable form for gaging the density of smoke or fog.

37 The numerous forms of appliances adapted for domestic

uses, all showed improved features, but need not be described here.

38 Other interesting exhibits included "Coalite" and other artificial smokeless fuels, spark and soot catching chimney cowls and special chimney construction for effecting a dilution of the smoke as it is emitted.

CONCLUSIONS ON SMOKE ABATEMENT

39 In judging of the merits of the various types of mechanical firing appliances exhibited, it should be recognized that all of these devices were designed for local conditions and for burning English or Scotch coals. These fuels cost from $2.50 to $3.50 per ton at the plant, contain from 30 per cent to 35 per cent volatile matter and 11,000 to 13,000 b.t.u. per lb. In these respects they correspond to our Illinois or other western coals. This class of fuel is not available in New England for economic reasons. The freight charges on coal in this section amount to about $3.00 or nearly three quarters of the average cost of fuel per ton. It is therefore not feasible or economical to burn any but the highest grade of bituminous coal in this part of the country for industrial purposes. This class of fuel should contain about 14,000 b.t.u. per lb. and not over 20 per cent volatile matter. On the chain grate, for example, which gives excellent results in England, this coal could not be burned satisfactorily. It would be impossible to maintain a proper ignition temperature with such a low per cent of volatile matter without an excessive draft which would give large losses to the stack. The chain grate is meeting with success in this country wherever fuel of the lower grades is available.

40 Of the other devices, perhaps the most important are those automatically controlling the air supply, both over and under the fire. These eliminate the personal factor of the fireman and should be effective on any type of hand-fired furnace.

41 In conclusion, smoke abatement may be best effected in the present state of the art of fuel burning by thorough consideration of the following conditions. Careful attention to one or all of these factors by competent engineers will increase the efficiency of any plant and with efficient combustion comes smokeless operation.

1 Selection of a suitable fuel with provision for maintaining same at fixed standard of heat value.

2 Careful scientific study of the conditions prevailing in the plant, including draft, composition of gases, temperatures, etc.
3 Design or selection of type of furnace or apparatus best adapted to meet these conditions.
4 Proper construction or installation of same.
5 Careful selection of operating force.
6 Suitable instrumental aids or guides for the firemen and responsible engineer.
7 Frequent and thorough inspection to insure maintenance of highest possible efficiency.

WORK OF THE HAMBURG SMOKE ABATEMENT SOCIETY

42 Among the delegates present at the London conference was Herr Nies, chief engineer of the Vereins für Feuerungsbetrieb und Rauchbekämpfung, of Hamburg, who later extended to the writer in Hamburg many courtesies which brought him into contact with the work of the society.

43 The Hamburg society, which was organized in 1902, is essentially different from other organizations of this character in the following important features:

a It is an entirely voluntary organization of manufacturers and other boiler users bound together only by a common desire to obtain greater efficiency and less smoke from their power plants.
b The working staff of the society is composed wholly of technically trained engineers.
c All boilers owned by members are under the systematic and scientific control of the expert staff of the society.
d All members benefit mutually by the published results of all tests and investigations made by the engineers.
e The economy of smokeless combustion is the cardinal principle and the members have been convinced of this in a most practical manner, in the savings effected in fuel consumption by their co-operative effort.

44 The society now numbers 436 members, representing 1381 boilers. The annual dues of members are small, being only $5 and $5 additional for each boiler in their plants. The society is entirely self-supporting and draws its funds from three sources, annual dues, payment for special work or reports, and payments for outside work.

45 The education and control of firemen in the proper performance of their duties are also undertaken by the fireman instructors on the staff of the society.

46 Comparative tests of fuels, smoke preventive devices, etc., are carried out by the engineers and the results are circulated among the members. Advice is also given on purchase of fuel, the buyers being educated by the engineering staff to appreciate the importance of this matter.

47 The engineering staff of the society consists of a chief engineer, four assistant engineers and eight trained firemen who act as fireman-instructors. For the chemical and thermal analysis of fuels, the society avails itself of the testing laboratory of Dr. Aufhauser at Hamburg. This laboratory, which is one of the most complete of its kind in Germany, tests about 2000 samples of fuel annually. Dr. Aufhauser also edits a periodical called " Smoke and Dust," to which the engineers of the society are frequent contributors.

48 The work of the Hamburg society is as practical and effective scientific work as is done anywhere along these lines. The efficiency of the plants, under its control, is materially bettered and maintained, the owners derive a substantial economic gain and the smoke problem is solved without the aid of the usual unsatisfactory legislation.

FOREIGN REVIEW

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FOREIGN REVIEW

The aim of the Foreign Review is to present, within the available space, the main data contained in the articles indexed. Where possible, reference is made to English or American publications containing fuller information on the subject treated. Measures are given both in original units and their English equivalents. In many instances, engravings and tables are reproduced. Opinions expressed are those of the reviewer, not of the Society. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *t* theoretical. Articles of exceptional merit are rated *A* by the reviewer.

Aeronautics

INFLUENCE OF A LATERAL WIND ON THE EFFICIENCY OF PROPELLERS (*Influence d'un vent latéral sur le rendement des hélices*, P James. *L'Aérophile*, vol. 20, no. 17, p. 397, September 1, 1912. *e*). The experiments of Hiram Maxim and especially those of Riabushinski have shown that the quality of a propeller is improved considerably when it is immersed in a current of air inclined to the axis of the propeller. In the experiments of Riabushinski a propeller in calm air gave, at 35 r.p.s., a thrust of 0.036, with a quality of 0.08. In a current of air normal to the axis these quantities increased regularly with the velocity of the wind, and with a wind of 6 m (19.6 ft.) per sec., the propeller gave, at 28 r.p.s., a thrust of 0.082, with a quality 1.16, or a thrust 2.3 times, and a quality 14.5 times, better than in calm air. Since the power required to produce a certain thrust is inversely proportional to the square root of the quality, it follows that in the second case only one-quarter of the power was required to produce the same thrust as in the first case. It must be noted, however, that these results have been obtained with a stationary propeller and therefore apply directly only to the case of elevating propellers. The author believes that this may explain why propellers located behind the planes have a better efficiency; it is because they work in the air thrown downwards by the planes, and therefore making a certain angle with their axes.

BICYCLE WITH WINGS (*Das geflügelte Fahrrad*, Jos. Hofmann, *Prometheus*, vol. 23, no. 1190, p. 721, August 17, 1912. 4 pp., 4 figs. *t*). Theoretical investigation of conditions of *motorless flight*. The author believed that under certain conditions which he investigates in detail, such competitions as the one recently held in Paris may be of value.

Researches on the Stability of Aeroplanes (Recherches sur la stabilité de l'aéroplane. *Bulletin de la classe des sciences, Académie Royale de Belgique*, J. Pacotte, 1912. No. 4, p. 292, 18 pp. *mt*). A theoretical investigation of the problem of *stability of aeroplanes*. The author's theory is based on the expression of the motions of an aeroplane by a system of nine differential equations. He introduces the notion of quasi-permanence, or a state tending towards permanence, and by this means limits the number of his equations to six, and further shows mathematically that a sufficiently steep ascent produces lateral instability, while a sufficient velocity produces lateral stability. The author investigates among other things the equation giving the criterion of stability; rolling, veering and pitching; necessary and sufficient conditions of stability. The whole subject is treated mathematically by elementary calculus in a very clear manner.

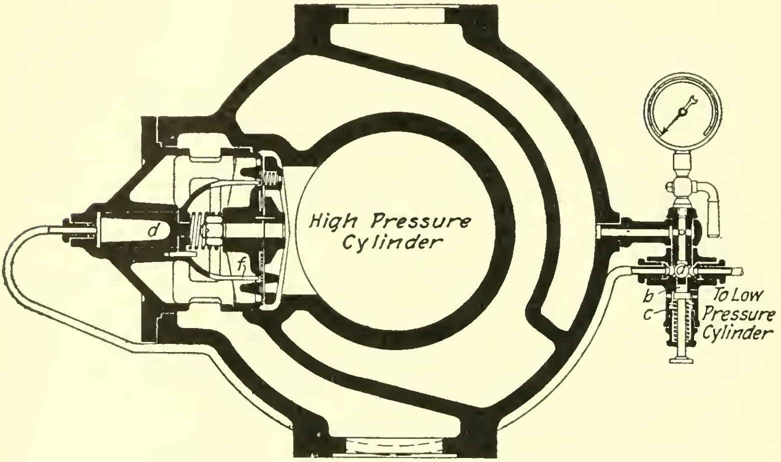

Fig. 1 Automatic Pressure Regulation of Air Compressor

Air Machinery

A Modern Swedish Compressed Air Plant (*Eine neuzeitliche schwedische Druckluftanlage*, H. Eriksson, *Glückauf*, vol. 48, no. 34, p. 1368, August 24, 1912. 4 pp., 8 figs. *hd*). Description of the compressed air plant of the Striberg Company. Rogler-Egnell valves are used practically the same as the Hoerbiger-Rogler valves described in The Journal by W. Trinks.[1] The pressure regulator of the air compressor is shown in Fig. 1. In the regulator valve three passages come together at a single point, the upper passage communicating with the compressed air discharge piping, while the other two are connected with devices for closing the suction valves of the cylinders. At the point of intersection of the three passages is placed a steel ball *a* held air tight against the opening of the upper passage by the spiral spring *c* pressing against the piston *b*. When the pressure of the compressed air on the ball exceeds the pressure of the spring, the ball is forced downwards, and compressed air, flowing into the two side passages, is led through small diameter copper pipes to the suction valves of the high and low-pressure cylinders where it acts on the piston *d;* the grip *f* forces the valve plate from its seat and opens the valve; the low-pressure cylinder is now connected with the external air, and the high-pressure cylinder with the piping leading to the intermediate cooler, while the open suction valve connects both sides of the piston, so that the compressor takes only as much power as when driven at no-load. When the air pressure has gone down enough to permit the spring to drive the ball back against the opening of the upper passage, the admission of compressed air to the piston *d* is cut off, and the compressor starts to work again. By adjusting the spring *c*, the regulator may be regulated for any desired pressure.

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[1] Reciprocating Blast-Furnace Blowing Engine, July, 1911, p. 775.

Hydraulics

HYDRAULIC TURBINE GOVERNOR (*Régulateur de turbines hydrauliques*. Le Génie Civil, vol. 61, no. 16, p. 324, August 17, 1912. 1 p., 3 figs. *d*). Description of a new type of *hydraulic turbine governor* built by Briegleb and Hansen, of Gotha, Germany. The object of the introduction of the new type was to make a governor that would be entirely independent of the dimensions of the turbine or head of water, and one which could therefore be manufactured in series. This governor, as shown in Fig. 2A, consists of two sets of parts divided by the dotted line *xx:* above is the governor proper, the parts of which are subject to but slight stresses; below the mechanism guiding the action of the turbine vanes and therefore developing or transmitting considerable mechanical efforts; as a consequence they are much stronger than those of the first group. The rotary oil pump *P* drives the oil from the tank *C* into a pipe provided with the by-pass valve *S* which lets the oil back into the tank *C* as soon as the pressure of oil in the air tank *R* exceeds a certain predetermined maximum. From this pipe the oil passes to the slide-valve which admits it either above or below the piston *A*, in its turn directly connected with the flood-gates of the turbine which it opens in going down, and closes in going up. The governor proper (upper part of the figure, above the line *xx*) consists of a sensitive centrifugal governor *B* which acts on the lever *D* having its other end connected to the slide-valve *T*. The axis of rotation of the lever *D* is a knife-edge at the top of the vertical rod *L*, the length of which may be adjusted by means of the casing *G*. The rod *L* is connected with another rod *M* by a nipple *F* provided with a friction pulley rolling along the surface of the friction gear *E* which is held against it by the spring *s* and kept in continuous rotation transmitted to it from the centrifugal governor spindle by means of the bevel gear shown to the left of *s*. The rod *M*, which is a prolongation of *L*, is connected with the stem of the piston *A* by the slot *H* which permits regulating the ratio of vertical motions of the rod *M* and piston *A*.

If, when the speed of the turbine is normal, the load should happen to fall off, the speed tends to increase, and the governor balls deflect. The lever *D* tips, with the end away from *B* going down, and forcing to its lower position the slide-valve *T* which opens to the oil under pressure a passage under the piston *A*, and sets the upper chamber of the cylinder in connection with the by-pass leading to the tank *C*. As a result the piston *A* rises, and thereby causes the flood-gates of the turbine to close. But at the same time this motion of the piston is transmitted, by the rods *M* and *L*, to the central fulcrum of the lever *D*, and causes it to bring the slide-valve back to its neutral position. Fig. 2B shows the lever in its various positions. The article describes also how the turbine may be allowed to run at no-load at a speed slightly different from that at normal load, which is convenient in the case of turbines driving alternators in parallel. The centrifugal governor is shown in detail in Fig. 2C. Its great sensitiveness comes from the elimination of the sleeve and a certain number of levers and fulcra which increase the friction. The essential parts of the governor are two movable masses *V* subject to the action of the centrifugal force and set at the ends of the levers *N*, the other ends of these levers being interconnected by the feed rod *B* while the spring *R* provided with knife-edge contacts (to decrease friction to a minimum) tends to counteract

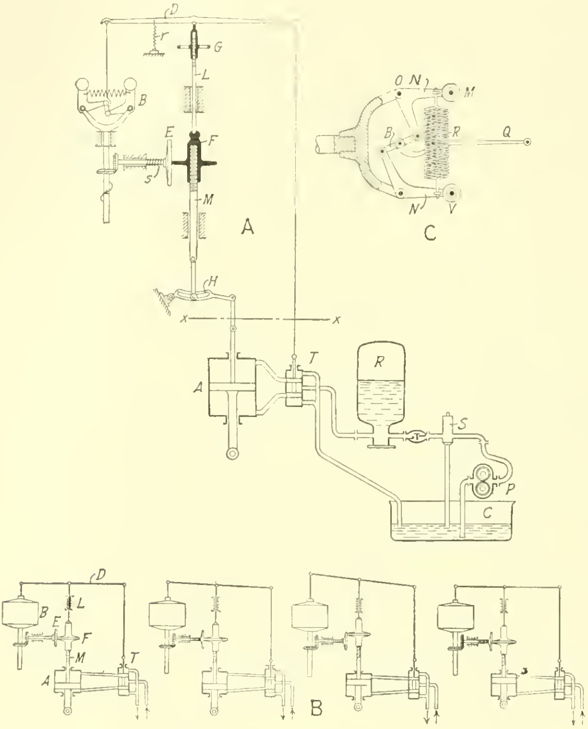

Fig. 2 Water Turbine Governor

the tendency of the balls V to deflect. The spindle transmitting the motion of the masses V to the governor valves has its point of application at the center of the feed rod B.

ON THE SELECTION OF A TYPE OF WATER TURBINE IN DOUBTFUL CASES (*Über die Wahl der Wasserturbinengattung in zweifelhaften Fällen*, F. Leubner. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 31, p. 1244, August 3, 1912. 2½ pp. *p*). When from the turbine engineer's point of view either a Francis or a free-jet turbine may be applied, and the economic speed of either is convenient for direct driving the selected type of generator, the installation of a Francis turbine instead of a free-jet turbine may be recommended only when the water to be used is clean, and no prolonged underloading of the turbine is expected during the operation. The author shows that if the speed of rotation of a Francis turbine is low, while the head is high and the amount of water large, the Francis turbine works practically as a pressure turbine; there is a tendency of the water to work loose from the back side of the blades, and eddies are formed which help the gases contained in the water to become free. These gases attack the blades, which in a short time become worn, and sometimes full of holes. The advantages of the Francis turbine, viz. large power output and low first cost, must therefore never be solely considered when selecting a turbine type.

EXPERIMENTS IN MAKING VISIBLE FLOW PHENOMENA IN SUCTION PIPING (*Versuche über die Sichtbarmachung der Strömungserscheinungen im Saugrohr*, V. Kaplan. *Die Turbine*, vol. 8, no. 22, p. 403, August 20, 1912. 4 pp., 12 figs. *e*). Data from experiments at the German Technical High School in Brunn, Austria. Glass suction pipes were used for the observation of various processes in the suction pipe of water turbines, and instantaneous photographs taken at various moments.

Processes of Filling and Emptying of the Suction Pipes. Immediately following the admission of the water into the upper penstock, it flows through the suction pipe in a spiral, due to the rotation of the wheel at no-load. When the pipe is half full, the photograph shows the formation at the bottom of the pipe of an almost uniform mass of liquid, still containing however eddying air-bubbles. The limiting surface of the water core is indicated by white foam above which the water is still in spirals. A few seconds later the pipe is filled and, if it has an air tight connection with the wheel frame, it appears to be filled with a perfectly clear and transparent liquid. The process of emptying the pipe is similar to that of filling.

The problem of *making visible flow phenomena* in suction pipes was attacked in several ways of which only one proved to be satisfactory. Introduction of clay or chalk powder caused the water to lose some of its transparency without clearly indicating the flow phenomena owing to the small size of the particles. Solutions of coloring matter proved unsatisfactory because they formed a thorough mixture with the water, and the "colored line" became indistinguishable. A better method was found by introducing some liquid insoluble in water, e. g. oil or petroleum, but it had the disadvantage of making the whole installation dirty. The author devised a method shown in Fig. 3 where *d* is a bunch of hemp introduced into the pipe on a piece of thin steel wire. The device indicated the flow phenomena very well when the hemp was not too long. The photographs in the article show the relation between speed of rotation, efficiency and direction of flow.

Internal Combustion Engines

New Experiments on the Combustion of Nitrogen in Exploding Gas Mixtures (*Neue Versuche über die Stickstoffverbrennung in explodierenden Gasgemischen*, F. Häusser. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 29, p. 1157, July 20, 1912. 7 pp., 11 figs. *eA*). The *combustion of nitrogen at high temperatures*, proceeding according to the simple equation $N_2 + O_2 = 2NO$, is considered to be a purely thermal reaction depending exclusively on the temperature, but independent of the manner in which the temperature is produced. This was further proved by the experiments of Nernst, who passed air through externally heated platinum or iridium tubes and found that the production of oxide of nitrogen varies with temperature, or follows the equation of van't Hoff,

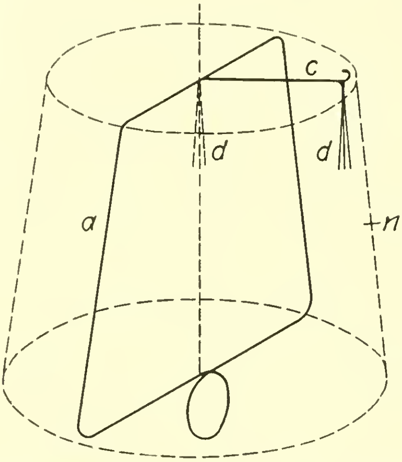

Fig. 3 Apparatus for making Phenomena of Flow in Water Pipes Visible

$$\frac{d \ln K}{dT} = -\frac{q}{RT^2}$$

where q is the heat change due to reaction, R gas content, T absolute temperature of transformation, K constant of equilibrium, which is in this case determined by the equation

$$K = \frac{(NO)}{(N - 0.5NO)(O - 0.5NO)}$$

with NO contents of oxide of nitrogen, N and O contents of nitrogen and oxygen apart from what is contained in the oxide of nitrogen, all in percentage of volume. The integration of the van't Hoff equation gives, with decimal logarithms instead of natural,

$$\log K + \frac{9424}{T} = 1.07$$

which expresses also the maximum amount of oxide of nitrogen which can be formed in a gas mixture heated to T deg. and containing any amounts of nitrogen and oxygen, provided the formation of oxide of nitrogen is a purely thermal one, and the chemical equilibrium corresponds to the temperature T and is constantly maintained so.

In a series of experiments the author found that the contents of oxide of nitrogen varies with the size of the explosion chamber: while in his first experiments with small bombs the contents of oxide of nitrogen corresponded fairly well to what would be theoretically expected, in the later experiments with three times larger bombs the amount of oxide of nitrogen produced was found to be 86 to 116 per cent above what was theoretically expected. The author analyses the possible reasons for such a divergence, and ascribes the production of the excess of oxide of nitrogen to *photochemical effects*, a supposition partly supported by the experiments of Warburg and Berthelot with the formation of oxide of nitrogen by silent electric discharges in air. One of the interesting things shown by these experiments is that as the time during which the temperature remains at its maximum increases, the contents of oxide of nitrogen increases also, apparently according to a parabolic law. For the application of the above data to the fixation of atmospheric nitrogen see Engineering, September 6, 1912, p. 329.

Investigation of Carbureters of Motor Vehicles (*Studie über die Vergaser von Motorfahrzeugen*, A. Heller. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 27, p. 1075, July 6, 1912. 9 pp., 19 figs. *teA*). Surface carbureters are now antiquated although the fact that they produce actual evaporation of the fuel, and therefore deliver a mixture with no fuel present in liquid form, is an incentive to return to that type. The spray carbureter was first theoretically investigated by Krebs who proposed his well-known formulae for the velocities of the flow of fuel and air through a carbureter nozzle. He proved that as the speed of rotation of the engine and the partial vacuum in the carbureter increases, there must be additional passages open to the air in order to keep the ratio of mixture between fuel vapor and air invariable. Krebs embodied his ideas in his carbureter known as the Panhard and Levassor, and nearly all the carbureters in use are based on the Krebs law of additional air supply. Still the theoretical bases of the Krebs law are not correct because the flow of fuel through a nozzle of a carbureter does not follow the law $v_B = \sqrt{2gh}$ or $v_B = \sqrt{2g(h-h')}$, but, as was shown by Rummel and others, rather a law corresponding to the Poiseuille's law.[1]

The theory of internal friction leads to the assumption of proportionality between the pressure of frictional resistance and the first degree of velocity, or

$$p_r = cv$$

The lack of agreement between this formula and experimental data is usually explained by the fact that while the theory of friction assumes a liquid jet of constant direction and close flow there are actually eddies of which the influence varies directly as the square of velocities. According to Reynolds the effect of eddies appears when the velocity exceeds a certain critical value (in c. g. s.)

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[1] For Poiseuille's law, see O. Reynolds, On the Motion of Water and the Law of Resistance in Parallel Channels, *Philosophical Transactions*, 1883, vol. 174, p. 935.

$$v_{\mathrm{k}} = 26\frac{1}{d}\frac{\eta'}{s^1}$$

where η' is the relative coefficient of internal friction, and s' the relative density of the fluid, both with respect to water of 10 deg. cent. (50 deg. fahr.). Reynolds found also that for the range above the critical point frictional resistance is proportional to the 1.7 power of the velocity. There is therefore in the region of the critical velocity a state of unstability with formation of eddies at the slightest disturbance. Further Grüneisen has shown that the value of the critical velocity at which values calculated from Poiseuille's formula differ from actual values by more than 0.1 per cent, is considerably lower, viz. (in c. g. s.):

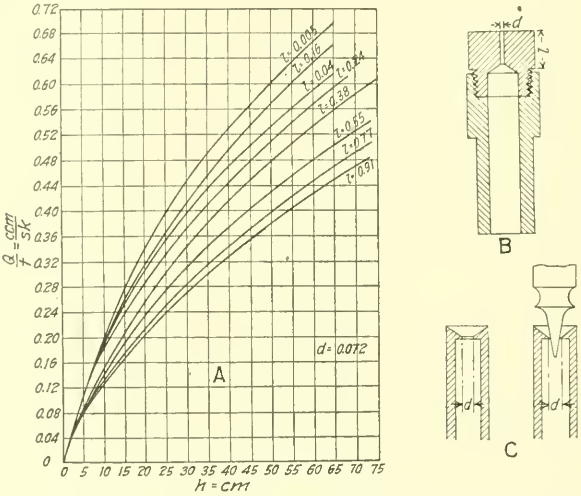

FIG. 4 *A* RUMMEL'S DATA ON FLOW OF WATER THROUGH CARBURETER NOZZLES; *B* AND *C* CARBURETER NOZZLES

$$v_{\mathrm{k}} = 6.6 \times 10^{-6}\left(\frac{\eta'}{s'}\right)\frac{1}{d}\left(\frac{l}{d} - 4.5\right)^{2.08}$$

and therefore one cannot assume any longer the existence of a perfect proportionality between the first degree of the velocity of efflux and frictional resistance.

Rummel assumes for the frictional resistance a law of the form

$$\frac{p_{\mathrm{r}}}{\gamma_{\mathrm{B}}} = a_1 v_{\mathrm{B}} + a_2 v_{\mathrm{B}}^2$$

the basis for this assumption being that, theoretically, only the first 2 deg. of velocity can affect frictional resistance. He uses for the flow of fuel through the nozzle of a carbureter the usual law of flow of liquids

$$\frac{v_B{}^2}{2g}+h'+\frac{p_r}{\gamma_B}=h_o-h$$

where $h_o=0$ (atmospheric pressure). This finally gives him the following equations:

$$c_1\,Q_B^2+C_2\,Q_B=H$$

$$c_1=\left(\frac{1}{2g}+a_2\right)\frac{1}{F_B{}^2}$$

$$C_2=a_1\frac{1}{F_B}$$

where Q_B is the amount of fuel flowing in time t, and F_B the cross-section of the nozzle outlet. Subscript B refers to fuel (*Brennstoff*), and L to air (*Luft*). The values of c_1 and c_2 have to be determined for each case. Rummel made his experiments with water; Fig. 4*A* gives the results of his experiments on the flow of water through carbureter nozzles, made with a nozzle of variable length and 0.072 cm. (0.028 in.) in diameter. It is clear that frictional resistance has the greatest influence on the amount of fuel delivered by a carbureter nozzle. The author quotes some experiments on the flow of fuel through carbureter nozzles which show that the above equations are approximately correct (Cp. *Horseless Age*, August 19 and 26, 1908).

The flow of air through carbureter openings may be accepted to follow Zeuner's equation of velocity:

$$v_L=44.4\sqrt{T\left(1-\left[\frac{p_0}{p}\right]^{\frac{m-1}{n}}\right)}$$

where T is absolute initial temperature, p_o initial or external pressure, p pressure at the nozzle, m outflow exponent equal to 1.286. Since the variations of pressure in the case of flow through a nozzle are extremely slight, the formula may be simplified by neglecting the changes in specific volumes, this being permissible in such cases as was shown by the experiments of Durley.[1] This brings us to the Fligner formula for the quantity discharged; for the average temperature t this formula may be expressed in the form

$$\frac{Q_1}{t}=a\,F_L\sqrt{h}$$

The processes of flow of air and liquid fuel through the carbureter now appear to be approximately clear. The relation between the amount of fuel and air discharged and the partial vacuum H in the carbureter is such that it alone may sufficiently explain why the richness of the mixture increases with the increase in the speed of the motor.

In actual design an effort is made to keep the relation $z=\frac{Q_B}{Q_L}$ as much as possible constant for all values of H. This can be done in the following ways:

(*a*) The carbureter is adjusted usually by suitably adjusting the nozzle opening on the testing stand, for a given minimum partial vacuum so that at this partial vacuum z may have what is considered to be the best value (say 1:16), and then larger air passages are provided in order that, as the speed of the motor rises, the minimum partial vacuum should not increase in proportion to the increase in the suction action of the motor. This is practically what the Krebs process does. Only automatic supplementary air valves may be used, since on the road the partial vacuum cannot be observed or measured so as to be adjusted by hand for various speeds. Though theoretically unattackable, the method of supplementary air admission in practice is far from being satisfactory. The adjusting and testing of carbureters, owing to the lack of proper methods of measurement, but seldom give sufficiently reliable results, and at the same time constitute a hindrance to mass production. It is inconvenient to have motors constructed on gages with no filing or scraping allowed, run for hours on the testing stand only in order to have the carbureter adjusted. The curvilinear shape of the supplementary air passages cannot be copied so exactly in factory production as to require no subsequent filing, and it is just as little possible to produce springs in large amounts for the automatically acting air valves in which the tension, when the spring had been placed in the assembled carbureter, should vary in exact accordance with requirements established experimentally. The supplementary air method gives therefore machines which work well on the testing stand, but not in the average conditions of street work.

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[1] Trans. Am. Soc. M. E., 1906, p. 193.

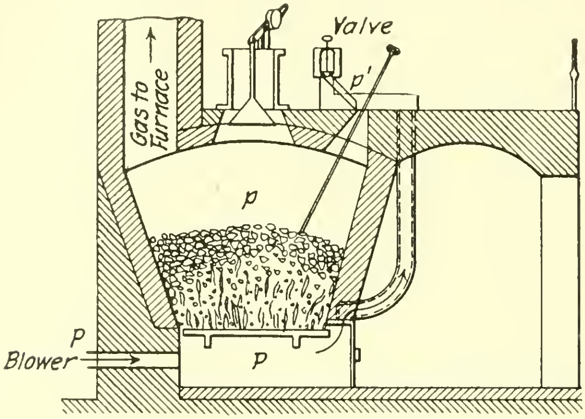

Fig. 5 Gas Producer with Pernot Regulator

(*b*) The second method of maintaining the ratio of mixture $z = \frac{Q_B}{Q_L}$ constant is by having the air admission opening of constant cross-section, and varying the admission of fuel. This method is comparatively little used, because, especially with fine nozzle openings and in the hands of inexperienced drivers, it may lead to excessive consumption of fuel. If this method be applied at all, there ought to be a device for contracting the fuel passage when the motor begins to run at high speed; in most of the so-called carbureters with adjustable valves (*Registervergasern*), however, when the partial vacuum increases, additional passages for fuel are opened (a full discussion of the Zenith carbureter follows).

The solution of the problem of constructing a carbureter that would work equally well at all partial vacua in the mixing chamber lies in finding a better form of nozzle. The usual carbureter nozzle is made generally with a cross-section like that shown in Fig. 4*B*, and to them applies Rummel's formula for the outflow

$$c_1Q_B^2 + c_2Q_B = H.$$

Should, however, the nozzle be designed that the simple parabolic law

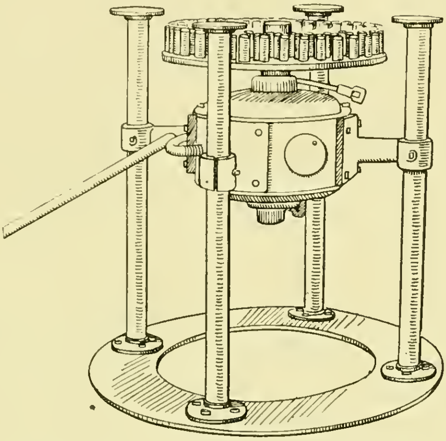

Fig. 6 Oerlikon Centrifugal Sand Cleaner

$cQ_B^2 = H$ might be applied to it, all the troubles of the present carbureter construction would be eliminated since the flow of air and liquid would follow the same law, and the ratio of mixture would be independent of the partial vacuum in the mixing chamber. The author proves experimentally that nozzles with a cross-section like that shown in Fig. 4C very closely approach this ideal condition in being nearly independent in their action of variations in partial vacuum. The peculiar characteristic of such nozzles is the extremely short length of their tubular part. In his work Rummel found that the regularity of the ratio of mixture may be improved by the use of very short nozzles. He, however, limited the length of nozzles to 1 mm (0.039 in.), but the shorter the nozzle, the less noticeable is the effect of capillarity which is the main cause of the flow of fuel differing from that of air at various partial vacua. It would therefore be reasonable to expect that in a nozzle with no capillarity at all, the flow would follow the parabolic law, just as in the case of flow through an opening in a thin wall. The article also contains data of tests on variation of pressure in the suction piping of a multi-cylinder engine.

REGULATOR FOR GAS PRODUCERS HAVING ARTIFICIAL DRAFT, C. PERNOT SYSTEM (*Régulateur de marche, système C. Pernot, pour gazogènes a tirage forcé*, A. Maillon. *La technique moderne*, vol. 5, no. 4, p. 137, August 15, 1912. ½ p., 1 fig. *d*). Description of a device intended to be used with gas producers having a draft produced by blowing in steam or air. If the pressure below the furnace be *P* (Fig. 5), there will be a pressure *p*, less than *P*, above the layer of coal. By means of a suitably arranged system of passages, a part of the fluid at the pressure *P* is conducted from under the furnace to an adjustable steam expander, whence it comes out with a pressure *P′* below *P*, but above *p*, and this fluid at a pressure *P′* is used for blowing in through the handholes while the attendant uses the slice-bar. If the pressure *P′* is conveniently chosen, it will prevent the gases at the pressure *p* from escaping from the furnace, which is important from a hygienic point of view. It is also important in regulating automatically the action of the producer. Suppose that the pressures *P*, *p* and *P′* have been so adjusted that there is no evolution of smoke or deleterious gases at the handholes. Should cracks appear in the fuel bed or should the bed become too thin, the ratio of pressures will change, and the furnaceman will have to see to it for his own convenience. The author states that this device was tried with satisfactory results with Siemens gas producers.

Machine Shops

NEW CENTRIFUGAL SAND CLEANER (*Neue Sandschleudermaschine*, H. Thieme. *Eisen-Zeitung*, vol. 33, no. 31, p. 556, August 3, 1912. 1 p., 3 figs. *d*). Description of a centrifugal sand cleaner lately placed on the market by the Oerlikon company. The sand is thrown from above into the hopper (Fig. 6), and falls upon a horizontal plate (Fig. 6) bolted to the end of the vertical shaft of the electromotor, and provided with three concentric rows of vertical staggered round bolts. The motor may work on direct current or alternating current, and take power from any lighting circuit. The sand is thrown by centrifugal force against the rows of bolts, forces its way through them and falls in a finely divided state in a circular heap on the ground ready for use. The iron particles stick in the middle of the plate and may be easily removed. The sand may be used for machine molding without further screening or other treatment. A 2-h.p. motor working at 960 r.p.m. can treat as much sand as two or three men can shovel into the hopper, or approximately 300 to 500 kg (660 to 1100 lb.) per minute. It does not require special foundations.

AUTOGENOUS WELDING OF COPPER AND COPPER ALLOYS (*Die autogene Schweissung von Kupfer und Kupferlegierungen. Acetylen*, vol. 15, no. 14, p. 125 (in the pagination of the appendix "*Autogene Metallbearbeitung.*" 5 pp., 5 figs. p). R. Amedéo, engineer of the Union de la Soudure Autogène in Paris has published a monograph on the above named subject in which he makes the following recommendations for the autogenous welding of copper: (*a*) the metal added (melt-bar) must have a melting point near that of the material welded; (*b*) it must absorb as little gas as possible, in order to produce a joint without porous places; (*c*) it must contain some deoxidizer capable of breaking up solutions of oxides; this deoxidizer must be such as easily to penetrate throughout the metal and, after reducing the oxides, form such combinations which would not tend to remain in the molten metal; the amount of the deoxidizer present must be just large enough to deoxidize the metal melted by the welding flame, since an excess of the deoxidizer is apt to affect the mechanical qualities of the metal. Mr. Amedéo also points out that the use of electrolytic copper is entirely unsatisfactory, since it forms weak joints owing to the absence of some action preventing the formation of oxides. The application of welding fluxes is not sufficient to destroy copper oxides, because their action is superficial, and the oxides remain in solution in the molten metal.

The article seems to give preference to phosphorus-copper alloy for added metal, but contains also a discussion of copper alloys with aluminum, zink, tin and antimony.

INSTALLATION FOR AUTOMATIC PREPARATION OF CORE SAND IN THE FOUNDRY OF DE FRIES & CO., IN DÜSSELDORF, GERMANY (*Automatische Kernsandaufbereitungs-Anlage in der Giesserei von De Fries & Cie., Düsseldorf. Giesserei-Zeitung*, vol. 9, no. 16, p. 516, August 15, 1912. 2 pp., 4 figs. *d*). Pure Rhine gravel with from 3 to 5 per cent of very viscous oil is used for making cores for radiators and sectional boilers; as long as hand preparation and mixing were used, the distribution of oil throughout the core was not uniform and when the core was dried, cracks would appear. This difficulty was overcome when the mixing machine shown in Fig. 7 was introduced. The gravel in carloads is placed in the bunkers *a*, and thence conveyed by the conveyer *b* to the bin *c* provided at the bottom with a telescoping tube. The other end of the tube leads to a rotating wiper plate *d* which throws on to the tub mixing machine *e* all the gravel falling outside of the telescoping tube. The oil is brought to the mixing machine by a wheel with buckets which takes a definite amount of oil from the tank and delivers it to the machine through a channel opening into the mixer. From the mixer the sand is taken up by the conveyers *f* and *g* and delivered to the bin *h* located above the core benches.

Mechanics

THE GYROSTATIC EFFECT AND ITS APPLICATIONS (L'effet gyrostatique et ses applications, W. Bogaert. *Bulletin technique de l'association des ingénieurs sortis de l'École Polytechnique de Bruxelles*, vol. 10, Nos. 7, 8, 9, pp. 305, 393, 489. May-July 1912. 240 pp. 76 figs. *mtpA*). Owing to lack of space only the last part of this series of articles can be abstracted here. The author deals with systems in motion required to follow a certain path, but at the same time exposed to parasitic motions, the elimination of which constitutes the problem of stability. The gyrostat possesses the property of resisting, in proportion to the speed of its rotation, all efforts tending to change its direction of action (*désorientation*), and is therefore widely used in stabilizers, of which there may be two kinds: (*a*) the stabilizer of a normally stable system (pendulum, ship) which has only to keep the amplitude of oscillations within certain limits, and (*b*) of a normally unstable system (bicycle, airship), where it must first establish a stable equilibrium, and then act as an oscillation absorber.

The stabilizer must therefore consist of (*a*) an apparatus sensitive to changes of orientation; (*b*) an element able to produce or absorb mechanical energy, and (*c*) a system of transmission between the two elements arranged in such a manner that the second should instantaneously develop the energy required to oppose any change of orientation indicated by the first. A gyrostat may be applied for stabilization in either of two ways: as an apparatus sensitive to changes of orientation, and directing the action of a servo-motor, or it may also act as an intermediary between the source of energy and the stabilized system, and itself carry the whole effort of stabilization. As an apparatus indicating changes of orientation it is preferable to a simple pendulum or liquid level, since in order that the indications of the latter be instantaneous, they must have a long period which would make their deviations slight for small variations of position, while the gyrostat is sensitive not only to the change of orientation, but to its velocity as well.

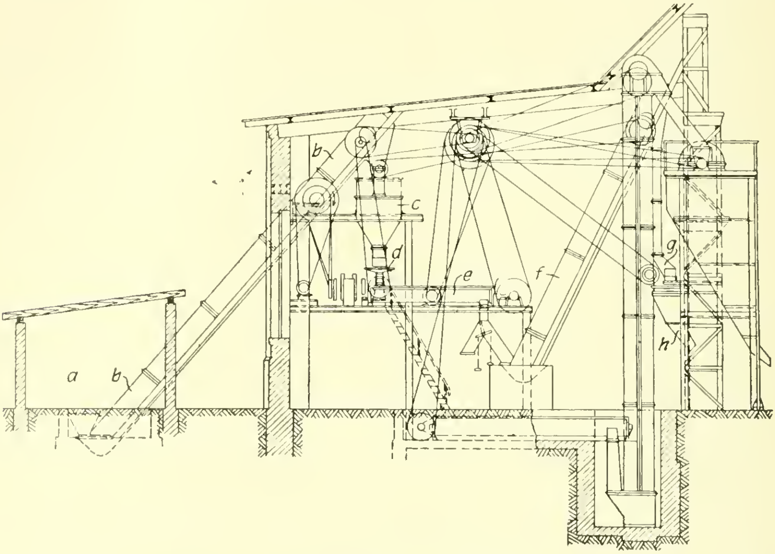

Fig. 7 Humboldt Arrangement for Automatic Preparation of Core Sand

The following rules must be applied when a gyrostat is used in stabilizers: (*a*) The axis of the tore must not be placed parallel to the axis of the oscillations to be reduced or prevented. (*b*) It must be given an average direction as far as possible normal to the axis of the undesirable oscillations. (*c*) If no invariable direction can be given to the axis of the tore, its variations must be limited in some way to prevent its deviation from normal to the axis of undesirable oscillations and thus from exceeding a certain definite limit, e.g. 45 deg.; otherwise the gyrostat may take a position parallel to the axis of oscillations and become insensitive.

Among apparatus of the first class, i.e. with the gyrostat only indicating changes of orientation, the author describes in some detail the Obry and Howell torpedo apparatus, and the Regnard airship stabilizer. The theory of the stabilizers of the second class is treated in detail, the author establishing the following theorems for the case when the axis of the gyrostat does not deviate much from the average position normal to the axis of oscillations of the system. (Owing to lack of space, the proofs of the theorems have to be omitted) : (*a*) A gyrostat taking part in a pendular motion has no effect on this motion if the position of its axis is invariable with reference to the system. (*b*) In the same case a gyrostat may introduce a stabilizing couple if the position of its axis is not invariable with reference to the system, i.e. if the gyrostat is capable of motions of precession; from this the author proves that an initially unstable pendulum may be rendered stable by the presence of a gyrostat having a kinetic moment greater than a certain definite value. (*c*) A gyrostat taking part in a pendular motion produces a stabilizing effect if the gyrostatic carriage, free as regards motions of precession, constitutes itself a compound pendulum. But the stabilizing action of the gyrostatic carriage acting as a pendulum is effective only when there is no friction.

As applications of the gyrostat as stabilizer, the author describes the ship stabilizer of Otto Schlick (including a discussion of the theory of ship rolling) and the monorail railway. The most valuable part of the article is the mathematical treatment of the theory of the gyrostat and its applications.

A list of books and articles on the gyrostat is appended to the article.

TORSIONAL OSCILLATIONS OF A ROD WITH FIXED AXIS OF ROTATION AND ANY DISTRIBUTION OF MASS SYMMETRICAL TO THE AXIS OF ROTATION, UNDER THE ACTION OF ANY HARMONIC FORCES (*Verdrehungsschwingungen eines Stabes mit fester Drehachse und beliebiger zur Drehachse symmetrischer Massenverteilung unter dem Einfluss beliebiger harmonischer Kräfte*, Professor Gümbel. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, nos. 26 and 27, pp. 1025 and 1085, 10 pp., 13 figs., *tA*). An interesting investigation passing in parts from the domain of mechanics to that of physics. The author shows that every elastic system rotatable about a definite axis and subject to the action of harmonic couples may be replaced, on condition of neglecting oscillations due to bending, by an elastic rod of zero mass and definite length with singular masses placed at certain distances from the axis of rotation. He investigates the motion of such a system under the action of periodic forces, and establishes a method for determining the variation in oscillations for each point at any frequency whatever. Three cases are fully investigated, and the connection between the spectrum and structure of matter discussed. The motion of an elastic system under the action of damping forces is investigated, and the application of the above theoretical considerations to technical problems discussed, particularly methods for preventing torsional oscillations. Such oscillations in the case of elastic systems of masses may be best prevented by avoiding the use of periodically varying torques. If this cannot be done, the influence of periodically varying torques decreases as the difference between the free period of vibrations of the elastic system of masses and the frequency of the torques increases. In the case of technical applications of torsional oscillations, e.g. in ships, motor shafts, etc., the rule should be adhered to: to keep the frequency of the driving force at least 15 to 25 per cent below that of the free vibrations, when the latter are of the first order (with a point of inflection on the oscillation curve). If the oscillations are of higher orders, the oscillation curves for different frequencies have to be determined, and the frequency of the driving force to be selected in such a manner as to keep the variation in oscillations within permissible limits. If the problem is not to eliminate oscillations entirely, but only to keep free from them a certain part of a system, the curve of oscillations may be so modified by the addition of suitably placed masses or introduction of supplementary forces as to make the variation in oscillations for a certain point equal zero. To do this, each case must be investigated separately. The location of the point of application of the driving force is of considerable importance. If the force is applied at a point on the rod or shaft where the amplitude of torsional oscillations is zero, such a force can transmit no work to the rod, and (since a certain amount of work is necessary to overcome internal or external damping forces), will neither cause new oscillations, nor increase those already present. On the other hand, the driving force is all the more liable to produce oscillations, the greater the amplitude of vibrations at the point of application of the force.

Pumps

CONTRIBUTIONS TO THE THEORY OF PLUNGER PUMPS (*Beiträge zur Theorie der Kolbenpumpen*, Karl Mayer. *Die Fördertechnik*, vol. 5, no. 7, p. 148, July 1912 4½ pp., 1 fig. *tA*.). In double-acting twin pumps with equal working piston areas the coefficient of cyclic variation, the maximum velocity of the water column under suction or pressure, and the acceleration resistance may be reduced by keying on the cranks to an angle of coupling less than 90 deg. Another way in which this can be done in a double acting pump is to reduce in one pump the area of the piston on the side away from the crosshead, and to increase in equal proportion the piston area of the side of the connecting rod in the other pump so as to have the working piston area on the connecting rod side of one pump equal to that on the other side of the other pump, and vice versa. The author proceeds to show that the coefficient of cyclic variation, the maximum velocity of the water column under suction or pressure, and the acceleration resistance may be still more reduced by a combined application of the two methods above described. This may be used also in the case of twin differential pumps as well as for pumps with single-acting plunger pistons, provided in the latter case each pair of pistons is connected in the middle by a crosshead. He also shows that in the case of double-acting twin pumps with equal working piston areas the same may be done still more effectively by a suitable choice of the angle of coupling which may be both larger and smaller than 90 deg.

Fig. 8 gives the time-velocity diagram for an angle of coupling greater than 90 deg. The phases 2 and 4 are congruous throughout, while phases 1 and 3 are only related in a certain close way to the crank guide motion; in case of a rotating crank gear the compensation for error members of the second degree increase the maximum value of phase 1, and reduce that of phase 2. These compensations for error members of the second degree may, however, be neglected altogether since they are represented by a simple sinusoid of small amplitude and having practically zero value in the middle of the phase where the phases 1 and 3 reach their maximum. In a book published by him under the same title as this article, the author has shown that the maximum value of the phases 2 and 4 is

$$\max C_{r2} = \max C_{r4} = \frac{(F+f\cos 2\pi\sim x)\;(1+k\cos 2\pi\sim x)}{\sqrt{k^2+2k\cos 2\pi\sim x+1}} + \frac{kf\sin^2 2\pi\sim x}{\sqrt{k^2+2k\cos 2\pi\sim x+1}} \quad [1]$$

where the ratio of the working piston areas is expressed by $k=\frac{f}{F}$. Should the maximum values of the phases 1, 2, and 4 be equal to each other, the following equation holds good:

$$\frac{(F+f\cos 2\pi\sim x)\;(1+k\cos 2\pi\sim x)+kf\sin^2 2\pi\sim x}{\sqrt{k^2+2k\cos 2\pi\sim x+1}} = 2f\sin\pi\sim x+\lambda f\sin 2\pi\sim x \ldots [2]$$

After [2] has been divided by F and the numerator of the fraction simplified, it becomes clear that the later is only a square of the denominator, and equation [2] may be put in the simpler form:

$$\sqrt{k^2+1+2k\cos 2\pi\sim x} = 2k\sin\pi\sim x\;(1+\lambda\cos\pi\sim x) \ldots\ldots\ldots\ldots [3]$$

Should the maximum values of phases 1 and 3 be also equal, then

$$2F\sin\pi\sim x-\lambda F\sin 2\pi\sim x = 2f\sin\pi\sim x+\lambda f\sin 2\pi\sim x \ldots\ldots\ldots\ldots [4]$$

which may easily be reduced to

$$\cos\pi\sim x = \frac{1}{\lambda}\cdot\frac{1-k}{1+k} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [5]$$

If the angular functions be eliminated from [3] and [5], the equation

$$(k-1)^2+\left(10\frac{1-k}{1+k}\right)^2 k = \left(\frac{4k}{1+k}\right)^2 \left\{1-\left(5\frac{1-k}{1+k}\right)^2\right\} \ldots\ldots\ldots\ldots [6]$$

is obtained containing only one unknown k, which may be found by some method of approximation, e. g. the Newtonian process (it cannot be found directly, because the equation is of higher than fourth degree). The value

$$k=0.75545$$

satisfies the equation closely enough. When k has been found, the value of the angle of coupling for the minimum of the maximum velocity may be found from equation [5]. It is $x=91$ deg. 42 minutes.

The improvement of the maximum velocity in per cent is

$$\eta = 100\left\{1-\frac{2k\{\sin\pi\sim x+\lambda\sin 2\pi\sim x\}}{(\sqrt{2}+\lambda)\,\dfrac{k+1}{2}}\right\} \ldots\ldots\ldots\ldots\ldots [7]$$

and with $\lambda=\frac{r}{l}=\frac{1}{5}$ the value $\eta=14.89$ per cent is obtained.

The improvement (*in per cent*) *when both methods are applied together is thus considerably greater than when either* the method of selecting the best angle of coupling (the improvement in this case has been found to be only 6½ per cent), or that of the best ratio of areas (improvement 10.52 per cent) *are applied separately.*

Coefficient of Variation of Raising of Water. The author proves mathematically that the coefficient of variation of the raising of water cannot be decreased by using the most suitable angle of coupling and ratio of sections on the basis of a comparison between the six intervals of variation in Fig. 8: the author speaks of six intervals of variation, because he proves that two intervals are nearly equal to zero respectively. He shows further that it is more advantageous to make the four largest intervals of variation equal to each other by equating the maxima values in all four phases. With $\lambda = \frac{1}{5}$, the numerical value of the improvement in per cent, with the values of k and x as above, is

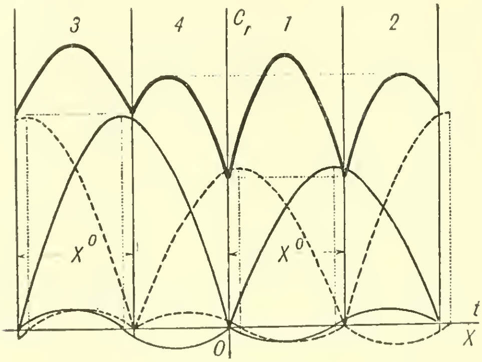

FIG. 8 TIME-VELOCITY DIAGRAM FOR PLUNGER PUMP WITH ANGLE OF COUPLING GREATER THAN 90 DEG.

$$\eta = 100 \left\{ 1 - \frac{0.75545 \times 0.6023}{0.87772 \times 0.61421} \right\} = 15.48 \ldots\ldots\ldots\ldots\ldots [8]$$

The improvement in per cent of the coefficient of variation in the case of equality of maxima values in all the four phases is smaller only by a fraction of a per cent than in the case of the sole application of the method of the best angle of coupling with equal working areas of pistons, but the combined method has the great advantage of securing two minima values at the same time, viz.: minimum of the maximum velocity of the suction or pressure water column and minimum of coefficient of variation of raising of water.

Minimum of Resistance of Acceleration. The author uses in this connection practically the same method as in the preceding paragraph. He shows first that the acceleration pressure at the beginning of the fourth, or second, phase cannot be equal, with the crank mechanism ratio $\lambda = \frac{r}{l}$ within practicable limits, to the pressure at the end of the phase, since if they were equal, the following equation would hold:

$$\begin{aligned} &f\ (1-\lambda)+F\cos 2\pi \sim x - F\lambda \cos 4\pi \sim x \\ &=F\ (1+\lambda)+f\cos 2\pi \sim x + f\lambda \cos 4\pi \sim x = 0 \quad\ldots\ldots\ldots\ldots [9] \end{aligned}$$

But this equation may be easily transformed into

$$1+\lambda\frac{1+k}{1-k}=\cos 2\pi \sim x-\lambda\frac{1+k}{1-k}\cos 4\pi \sim x \quad\ldots\ldots\ldots\ldots [10]$$

and that cannot be true as long as $\lambda = \frac{l}{r}$ is a rational fraction, because both $\cos 2\pi \sim x$ and $\cos 4\pi \sim x$ are less than unity and are negative, the angle of coupling being greater than 90 deg. The eight values of the maximum mass pressure cannot therefore be equal, and not more than six of them can be equal to each other. The author, as in the previous paragraph, proceeds to calculate the improvement, in per cent, in resistance to acceleration due to the use of the most favorable angle of coupling (x = 107 deg. 33 min.), and thus shows that the decrease of the resistance to acceleration due to the combined method is not larger than when either of these methods is applied separately. With a ratio $\lambda = \frac{r}{l} = \frac{1}{5}$, the acceleration pressure is the same as in a double-acting twin pump with working areas of pistons $\frac{f+F}{2}$ if the crank mechanism ratio is $\lambda = \frac{1}{10}$ and the angle of coupling of the pumps is 90 deg. The application of the combined method does not therefore affect the influence of the crank mechanism ratio on the resistance to acceleration any more than the application of either of the two methods separately. The acceleration pressure is nearly the same as when one of the two, the most favorable angle of coupling or ratio of areas, is selected. The article is to be continued.

Experiments of the Firm A. Borsig in Tegel near Berlin with Duplex-Spring Valves in High-Speed Water Pumps (*Versuche der Firma A. Borsig in Tegel bei Berlin mit Doppelfederventilen bei einer schnellgehenden Wasserpumpe*, O. Klepal. *Die Fördertechnik*, vol. 5, no. 8, 169, August 1912. 8 pp., 5 figs. *eA*). The author's fundamental contention is that for a steady and free-from-shocks operation an automatic valve must be loaded negatively, i. e. in the direction from the seat. The experiments were made with a differential pump having 60/85 mm (1.96/2.78 in.) diameter of the plunger piston, 200 mm (6.56 in.) stroke, with the suction and pressure air vessels arranged as shown in Fig. 9. The pump was driven by an electric motor provided with a device for varying the speed of rotation rapidly. Fig. 9 shows the construction of the suction and pressure valves which are both of the same design, viz. double-seat valves with two springs, of which one presses the valve on to its seat and the other lifts it up from the seat (in the rest of the article the pressure spring will be denoted by "+spring," and the raising spring by "—spring," the data referring to the first being denoted by subscripts d and those referring to the second by subscripts s.)

Owing to lack of space only data of the experiments which appeared to be the most characteristic have been reproduced here. The original numbers of the experiments have been retained to facilitate reference to the original.

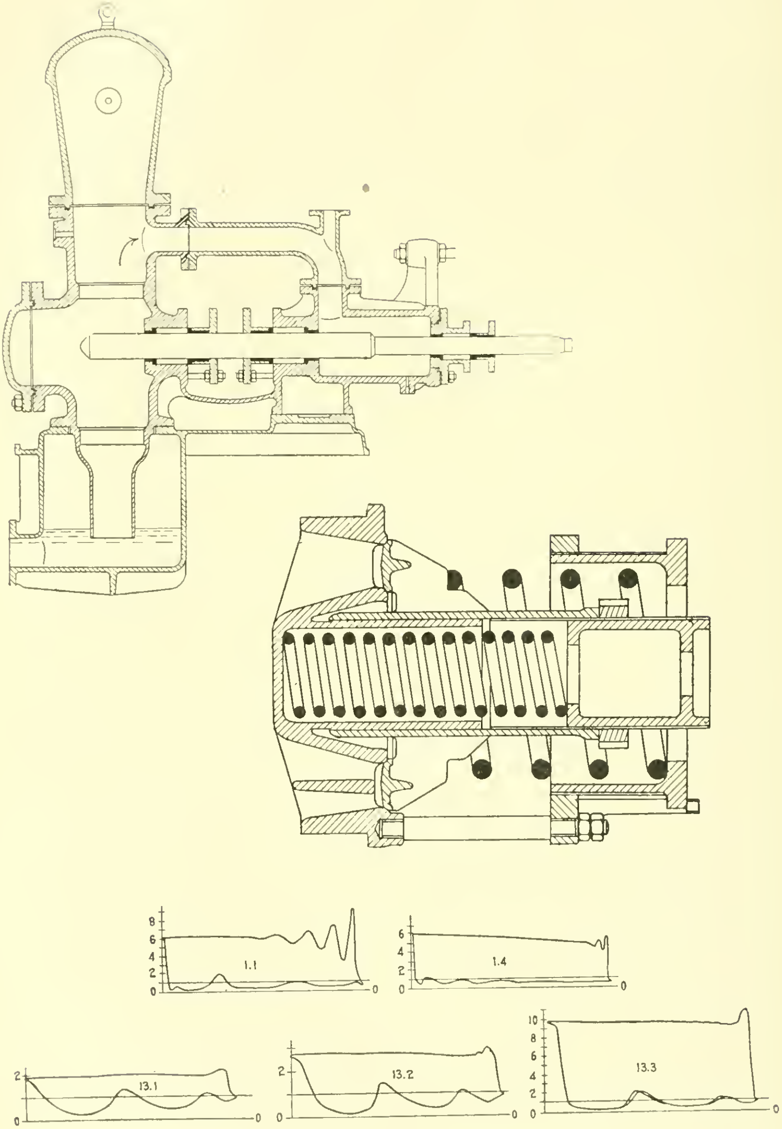

Fig. 9 High-Speed Water Pump with Its Valves, and Indicator Diagrams of Same

In experiment 1 both valves have negative loads, the pressure valve —7.79 kg, and the suction valve —4.95 kg. The two indicator diagrams are interesting in that they show how the indicator mass affects the form of the diagram, particularly that of the suction and pressure valves at the opening of the valve: the oscillations decrease with the decrease of the number of revolutions.

Experiment 11: Pressure valve negative load —7.86 kg, suction valve positive +30.8 kg; a Maihak indicator was used, with the result that the oscillations in the suction and pressure lines are smaller than when heavier indicators were used.

Experiment 13: Both valves are negatively loaded, —19.3 kg pressure valve, and —6.93 suction valve. This test shows clearer than any other one that it is possible to put a very large negative load on the valves without materially affecting the efficiency of the pump. At low pressures the suction line goes down, but improves at higher pressures, from 8 kg on. These diagrams show among other things that for each working pressure there must be a suitable load on the valves.

The highest speed was 330 r.p.m. at 9 atmospheres pressure. With suitably loaded valves the action was extremely regular, and only a very light metallic sound could be heard when standing behind the pump, but an observer standing at the crank bearing could not hear even that. Although the pump worked with 9 atmospheres pressure, it gave the impression of a machine working at no-load. The author states that other arrangements have been tried, but with none was it possible to reach a speed above 150 r.p.m., and the possibility of speeding the pump up to 330 r.p.m. with an arrangement as simple as this is certainly worth noticing. These tests have shown: that the pump refused to work whenever both valves were loaded positively; that the load on one valve affects the action of the other, and that the theoretical investigations which have treated the suction and pressure valves from the same standpoint, are incorrect; the speed of rotation as well as the working pressure affect materially the output of the pump and form of the indicator diagram.

Safety Appliances

ANNUAL REPORT OF THE TECHNICAL INSPECTOR OF SECTION IV OF THE "KNAPPSCHAFT-BERUFSGENOSSENSCHAFT" FOR THE YEAR 1911 (*Jahresbericht des technischen Aufsichtsbeamten der Sektion IV der Knappschafts-Berufsgenossenschaft für das Jahr 1911. Braunkohle*, vol. 11, no. 20, p. 313, August 16, 1912. 3 pp., *g*). The report contains, among other things, the description of the following noteworthy *safety appliances;* (*a*) apparatus for preventing bursting of flywheels on briquetting presses: when the flywheel speed exceeds a certain predetermined number of revolutions, a weight is so far projected by centrifugal force as to strike and release a lever, automatically closing the throttle valve and cutting off the steam admission; (*b*) under conveyor belts, just before the belt touches the roller with its hauling side, cleaning brushes are set rotating in a direction opposite to the motion of the belt. The rollers are therefore always clean, there is no further need to clean them while the machinery is in operation, and the numerous cases of the cleaners having their hands or fingers drawn in between the belt and roller are avoided; (*c*) the foot operating lever of a compressed air hammer is protected by a sheet-iron plate which protects the hammer from being set into operation by a piece of metal falling from the anvil and striking it; a case like this occurred in the practice of the inspector, and the installation of such a cheap and simple protective device is recommended wherever the foot lever is used on steam and compressed air hammers; (*d*) device for protecting laborers cleaning manholes leading to drying ovens in briquette factories from being hurt by feeding of the fuel: a two-part iron cylinder is placed over the opening, with a grate at the bottom nearly flush with the floor; the laborer stands on the grate, and is protected from falling coal by the cylinder.

TABLE 1 DATA SHEET OF ZOELLY TURBINE TESTS AT DÜSSELDORF, GERMANY

	November 1, 1910			December 29, 1911		
No. of test	1	2	3	1	2	3
Duration of test, min	70	54	57	61	120	60
R.p.m.	3000	3000	3000	3000	3054	3036
Useful output at terminals, kw	277	541	781	1047	1044	547
Efficiency of generator	73.6	84.6	88.9	91.0	91	84.6
Pressure in front of turbine valve, atmosphere absolute	14.3	14.15	13.78	13.71	12.98	13.1
Temperature in front of turbine valve, deg. cent.	263	282	291.5	304	249	237
deg. fahr.	505.4	539.6	556	579	480	458.6
Pressure in turbine exhaust connecting branch, atmospheres absolute	0.043	0.052	0.065	0.079	0.083	0.057
Total consumption of steam per hour, kg.	2764	4346	5825	7199	7867	4770
lb.	6070	9561	12815	15838	17307	10495
Consumption of steam per useful kw-hr., kg.	9.91	8.03	7.46	6.87	7.51	8.71
lb.	21.8	17.8	16.5	15.2	16.6	19.4
Correction for steam temperature,[1] per cent	−12.4	−9.5	−8.0	−6.9	−14.4	−16.1
Correction for vacuum,[2] per cent	+3.2	+3.2	+3.2	+3.2	+2.5	+2.5
Consumption of steam per kw-hr. at 350 deg. cent. (662 deg. fahr.) and 0.1 atmospheres absolute at full load, kg.	9.11	7.53	7.10	6.62	6.62	7.5
lb.	20.05	16.6	15.6	14.6	14.6	16.6

[1] 7 deg. cent. = 1 per cent.

[2] 1 per cent vacuum = 1.5 per cent steam consumption.

Steam Engineering

Results of Tests of Zoelly Turbines (*Versuchsergebnisse an Zoelly-Turbinen*, L. Grabau. *Zeits. für das gesamte Turbinenwesen*, vol. 9, no. 23, p. 363, August 20, 1912. 1 p., 2 figs. *c*). At the plant of the Düsseldorf Tube Company, Düsseldorf, Germany, two *Zoelly turbines* were installed in August 1910 by Escher Wyss & Co., of Zurich, with a capacity of 1500 h.p. each, designed for a pressure of 12 atmospheres, superheat 350 deg. cent. (662 deg. fahr.), and 90 per cent vacuum at full load; each turbine has surface condensation with rotary air and circulation pumps, and repeated cooling of cooling water. Each turbine drives a 550-volt alternating-current generator, with exciter.

Two series of tests were made: the first about 3 months after the two units went into operation and the second 13 months later. During that time the turbines worked about 4,000 hours, and delivered 1,450,000 kw-hrs. The purpose of the second series of tests was to find out whether or not there was any change in steam consumption during the above period: no such change has been found. Test data will be found in Table I. The consumption of steam of 6.87 kg. (15.2 lb.) per kw-hr. was at 92.1 per cent vacuum and 304 deg. cent. (579 deg. fahr.). Reduced to 350 deg. cent. (662 deg. fahr.) superheat, and 96 per cent vacuum, it would give a consumption of 6.01 kg. (13.2 lb.) per kw-hr. at full load of 1050 kw.

MODERN SETTING OF STEAM BOILERS (*Moderne Dampfkesseleinmauerung*, Pradel. *Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 35, no. 35, p. 365, August 30, 1912. 3 pp., 7 figs. *dg*). A general description in a somewhat popular style of modern setting of steam boilers.

PLANT FOR COALING BOILERS AT THE MINE ZOLLERN II OF THE GELSENKIRCH MINING COMPANY (*Die Kesselbekohlanlage der Zeche Zollern II der Gelsenkirchener Bergwerks-A.-G.*, A Pietrkowski, *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 29, p. 1164. July 20, 1912. 4 pp., 12 figs. *d*). Description of the *electric telpher line* used at the above plant *for conveying coal* from the washers to the boiler plant. The plant is an electric generating station capable of producing 12,500 kw. located at the coal mine of the company and supplying light and power to the surrounding communities and the city of Dortmund. About 100 tons of coal have to be delivered to the boilers daily. This is done mechanically by means of the described installation with an endless track lying at the elevation of the coal bunkers. The telpher cars are loaded either with nut coal from the washers or with rough coal; for the latter purpose they are lowered by special elevators to the loading openings located 6 m (19.68 ft.) below the level of the rails. The plant is designed to handle 35 tons per hour, and requires the attendance of only one man.

RESISTANCE TO FLOW IN THE DISTRIBUTING VALVES OF A RECIPROCATING STEAM ENGINE (*Strömungswideständе in den Steuerungventilen einer Kolbendampfmaschine*, E. Heinrich, *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 30, p. 1191, July 27, 1912. 8 pp., 15 figs. *e*). Description of, and data from, experiments having for their purpose the determination, by means of indicator diagrams, of the *coefficient of flow in the admission and exhaust valves* of a single-cylinder condensing engine made at the engineering laboratory of the Royal Technical High School at Stuttgart, Germany. Only the final conclusions of this investigation can be given here. The coefficient of flow (or the ratio between the amount of steam actually flowing through a valve at any time during the process of flow to the instantaneous value of flow, or the amount of steam that would flow through the valve if its entire cross-section were utilized and if there were no resistances) increases with drop of pressure. The coefficient of flow considered as the ratio between the values stated above, but taken from the beginning of the process of flow to the moment of observation (integrated value) may be taken for the admission valve to be approximately constant and equal to 0.55 for the admission curve beyond the dead point; for the exhaust valve it varies from 0.8 at the beginning to 0.62 at the end of the exhaust curve.

Strength of Materials and Materials of Construction

INVESTIGATION OF THE GERMAN SOCIETY OF BRIDGE AND IRON STRUCTURE BUILDERS OF IRON CONSTRUCTIONS USED IN BRIDGE BUILDING (*Untersuchungen des Vereines deutscher Brücken—und Eisenbaufabriken mit Eisenkonstruktionen für den Brückenbau*, B. Stock, *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 28, p. 1101, July 13, 1912. 9 pp., 31 figs. *eA*). The German Society of Bridge and Iron Structure Builders has been for many years carrying on a carefully planned series of investigations on *iron structures used in bridge building* through a special board formed of representatives of the Society, and the German ministries of Public Works, Education and Navy. The present article gives the main data of the third series of tests made under the direction of the board at the Royal Testing Laboratory at Gross-Lichterfelde West, viz.: (*a*) tension tests of riveted joints for the determination of the influence of the *process of riveting* (hand riveting, air riveting, toggle joint riveting with compressed air) on sliding resistance and breaking strength of the riveted joint; (*b*) tests on the *sliding resistance* in connecting plates with various forms of riveting; (*c*) tests on the influence of reduction of area on the tensile strength of *flat iron and angles;* (*d*) tests on bending of *angle webs.* All the rivets under tests were made without countersinking because it was found from previous experiments (1909) that *countersinking* in no way improves the qualities of the joint.

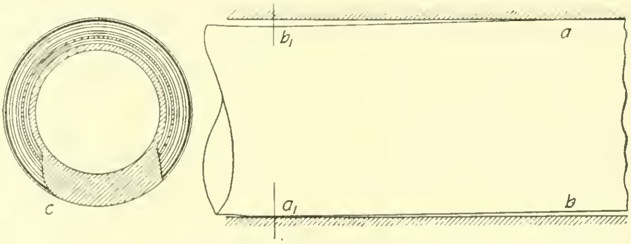

FIG. 10 YEAR-RINGS ON A LEAD PIPE

YEAR RINGS ON A LEAD PIPE (*Jahresringe an einem Bleirohr*, Kohlmeyer. *Chemiker-Zeitung*, vol. 36, no. 105, p. 993, August 31, 1912. 1 p., 1 fig. *d*). In making improvements in the water piping in the park of the Technical High School at Charlottenburg there was uncovered a lead pipe which had lain in the ground for 24 years at a depth of about 1 m (say 3.3 ft.). Part of the pipe lay in the earth unprotected, while one of the ends was imbedded in a cement foundation. The part which lay in the earth was found to be in a practically perfect state of preservation, but the part imbedded in cement underwent a rather peculiar transformation. As shown in Fig. 10B, the part of the pipe in the cement was slightly bent, and the contact with the cement was not uniform. It was found that just where the pipe entered into the cement, the lead was transformed into a compact, stonelike, yellow-red mass, and when this was knocked off near a part of the pipe where the lead was not affected, the latter had a scarred surface with silver white lustre, unusual for lead, which rapidly changed to a dull gray in air. The pipe was sawed through at b_1a_1 and the section polished (Fig. 10A). The pipe was 18 mm (0.7 in.) in diameter, with the walls 5 mm (0.196 in.) thick; at b_1 the yellow-red mass was 4 mm (0.15 in.) thick, but its thickness decreased gradually toward a_1. The most remarkable fact, however, was that on closer investigation it was found that the transformed mass consisted of concentric alternating yellow and red layers, 24 red and 23 yellow, or practically *one pair for each year the pipe lay in the cement*: the layers of one color were evidently formed during the wet, and those of the other color during the dry part of the year. An analysis proved that the colored substance was lead oxide, the yellow part being PbO, and the red Pb_3O_4. The fact that only the part lying in cement was so affected, appears to show that this peculiar oxidation of lead was due to the action of either the cement or cement water. The author undertook

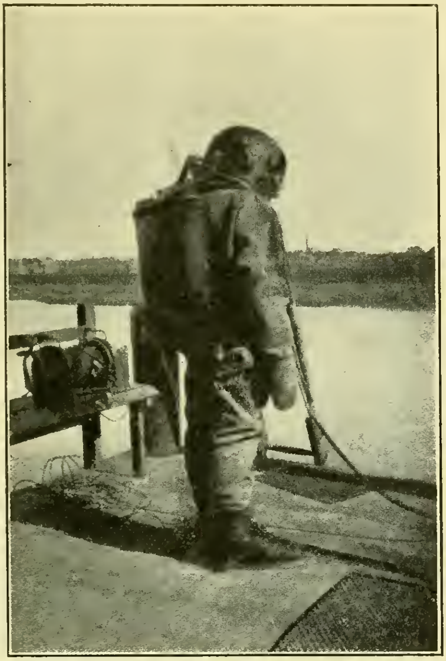

Fig. 11 Lubeck Diving Apparatus

a series of tests in this connection, of which he promises to publish the results in a few years.

Miscellanea

Something about Diving (*Etwas von der Taucherei. Prometheus*, vol. 23, no. 1188 and 1189, p. 689 and 713, August 3 and 10, 1912. 8 pp., 12 figs. *hd*). Description of various types of diving apparatus, from the Smeaton diving bell to that used at the present time. An interesting type is the Lubeck *diving apparatus without outside air supply.* The diver is provided with a cylinder filled with compressed oxygen besides the usual compressed air cylinder. When the apparatus is in operation, the oxygen flows through a pressure-reducing valve into a nozzle, sucks the used air through a hose (Fig. 11 at the left), and forces it through a vessel filled with anhydrous potash which absorbs almost entirely the moisture and carbon dioxide contained in the air; it returns the air through the hose at the right into the diver's helm supplied with oxygen and fit for breathing. The air supply of the apparatus is in continual circulation, and the oxygen used up is replaced by new. The diver receives 2 liters (0.76 cu. ft.) of oxygen per minute, and 60 liters (22.8 cu. ft.) of air. The oxygen and potash tanks are so proportioned that the diver is provided for two hours' stay under water; a lever at the belt allows him to regulate the consumption of air and oxygen. The inside pressure is regulated by an automatic valve placed in front at the shoulder which lets out the air whenever the inside pressure exceeds the water pressure by a certain small amount. The diver can decrease the inside pressure or blow out his suit by the air from the compressed air tank as he wishes, a considerable convenience in work at varying depths. The absence of an air hose is said to make this apparatus especially convenient for deep working.

The second part of the article is of a historical character, and contains descriptions of diving bells of the older types.

Industrial Mechanical Treatment of Stone (*Le travail mécanique de la pierre dans l'industrie*, J. Escard. *Revue de mécanique*, vol. 31, no. 1, p. 5, July 31, 1912. 29 pp., 44 figs. *d*). First of a series of articles on machinery used in the working of stone. The present article describes in detail various *saws for cutting stone:* reciprocating and band saws, metal and diamond, with fixed and movable teeth.

Some Special Tests of Automobile Carriages (*Quelques essais spéciaux sur les voitures automobiles*, Boyer-Guillon. *Revue de mécanique*, vol. 31, no. 1, p. 30, July 31, 1912. 25 pp., 15 figs. *cA*). Description of a very simple and convenient apparatus used for testing automobiles by the testing laboratory of the Conservatoire National des Arts et Métiers, and of tests on automobile suspensions and shock-absorbers.

Supplementary References

Miller for Torpedo Propeller Blades (*Foreign Review*, September 1912, p. 1390). Detailed description in *American Machinist*, vol. 37, no. 11, p. 457, September 12, 1912.

The Gyrostatic Effect and its Applications (p. 1571). Cp. Herbert Chatley, *Practical Gyrostatic Balancing*, London, 1912, particularly p. 16 ff.

GAS POWER SECTION

PRELIMINARY REPORT OF LITERATURE COMMITTEE

(XXI)

ARTICLES IN PERIODICALS [1]

ABWÄRMEVERWERTUNG BEI VERBRENNUNGSKRAFTMASCHINEN, K. Kutzbach. *Stahl und Eisen*, July 11, 1912. 2 ⅔ pp., 4 figs., 2 curves. *dp*.

Utilization of waste heat from internal-combustion engines.

ELYRIA OIL ENGINE. *Power*, August 27, 1912. 1⅓ pp., 3 figs. *d*.

Describes a two-stroke cycle oil engine of the Diesel type.

ENGINE, 400-B.H.P. TWO-CYCLE GAS. *Engineering*, August 30, 1912. 2 pp., 6 figs. *d*.

Describes a single-cylinder engine without any side shafts or eccentric gearing.

EXHAUST GAS TURBINE, AN. *The Engineer* (London), August 23, 1912. 2 pp., 4 figs., 1 table. *depA*.

Gas turbine connected to exhaust of petrol motor; theory and design of turbine by Mr. Montague Churchill-Shann, London.

EUROPEAN GAS ENGINE PLANTS, SOME RECENT, A. G. Lescarboura. *Power*, August 27, 1912. 1⅓ pp., 3 figs. *d*.

INTERNAL-COMBUSTION ENGINES, AMERICAN PRACTICE IN RATING, Parts 1, 2, 3, 4. T. C. Ulbricht and C. E. Torrance, Jr. *Power*, July 23, 30, August 6 and 13, 1912. 8½ pp., 9 tables, 36 curves. *em*.

This series of articles is the result of a thesis investigation made under the direction of Profs. C. F. Hirshfield and H. Diederiches of Cornell University.

JUNKERS OIL ENGINE, THE, J. B. Baker. *The Iron Age*, August 1, 1912. 2 pp., 8 figs., 1 curve.

Valveless engine with two pistons per cylinder for utilizing cheapest grades of fuel.

MARINE OIL ENGINE, 10-H.P. "MONOBLOC." *Engineering*, August 16, 1912. 2 pp., 3 figs. *d*.

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[1] Opinions expressed are those of the reviewer, not of the Society. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *h* historical; *m* mathematical; *p* practical. A rating is occasionally given by the reviewer, as *A*, *B*, *C*. The first installment was given in The Journal for May 1910.

Moteurs à combustion, Étude Theorique du Rendement Thermique des, T. Mancy. *Revue de mécanique,* June 20, 1912. 3 pp., 5 curves.

Theoretical study of thermal operation in internal-combustion engines.

Motor Ship Monte Penedo, The. *The Engineer* (London), August 30, 1912. 1 p., 2 figs., 1 table. *dpB.*

The ship, which is 350 ft. long by 50 ft. beam, is equipped with two Sulzer two-cycle Diesel engines; 1700 combined h.p. at 160 r.p.m.; 0.46 lb. Texas oil consumption per b.h.p.

NECROLOGY

E. HARRISON SYMINGTON

E. Harrison Symington was born in Baltimore, Md., March 1, 1879, and was educated in the public schools there and in the technical training school at Bethlehem, Pa. He entered Lehigh University in the class of 1898 and was graduated with the degree of M.E. His first experience was in the shop of the Lehigh Valley Railroad, with which company he remained till he was made engineer of tests, serving in that capacity for one year. In 1897 he entered the employ of Kilborne & Jacob Manufacturing Company of Columbus, Ohio, where he was in charge of the engineering and shop practice. He resigned this position to become manager of western sales of the T. H. Symington Company, and subsequently of their foreign business. When the concern opened its new malleable foundry in Rochester, N. Y., 1909, he was made works sales manager. His last years were spent as mechanical expert for the company, especially devising improved methods for draft gear designs for freight equipment.

Mr. Symington died in Baltimore, September 5, 1912.

FREDERICK J. MAYER

Frederick J. Mayer was born in Bremerhaven, Germany, July 4, 1853. He received his technical education at the University of Stuttgart, and came to this country in 1870, after having seen active service in the Franco-Prussian War. He engaged with the Bartlett Hayward Company of Baltimore, Md., as draftsman, and while with them, as chief engineer, he designed, erected and operated many of the large gas plants constructed by the company. Among them may be mentioned the Consumers' Gas Fuel and Light Company of Chicago, in 1883, and the Nassau Gas Company's water gas plant, Brooklyn, in 1891. In 1906 Mr. Mayer severed his connection with the Bartlett Hayward Company and organized and established the Didier-March Company in the United States, taking the position of general manager.

Having made a life study of the carbonization of coal for metallurgical and domestic use, to him must be credited the establishing of some of the best foreign apparatus for such purposes in the United States by both the intermittent and continuous system in vertical retorts for gas-house practice. He not only secured these rights for the United States, but promoted the companies, erected the plants and put them into successful operation. Mr. Mayer had invented many improvements for the manufacture of illuminating gas, patents of which were secured in the United States, Germany, France, and England.

At the time of his death, September 4, 1912, he was still an ardent and tireless worker, well known in this country and abroad. He was a member of the American Gas Institute, the Illuminating Engineering Society, Chemists Club, American Peat Society and many other technical organizations.

EMPLOYMENT BULLETIN

The Society considers it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is pleased to receive requests both for positions and for men. Notices are not repeated except upon special request. Names and records, however, are kept on current office list three months, and if desired must be renewed at the end of such period. Copy for the Bulletin must be in hand before the 12th of the month. The list of "men available" is made up from members of the Society, with names of other good men not members, and are on file in the Society office. Information will be sent upon application.

POSITIONS AVAILABLE

0194 Note: Published in the September Journal, address should read, Estate of M. Shaik Adam Ipoh, Straits Settlements, or send care of A.S.M.E.

0195 Efficiency engineer for concern in Middle West manufacturing agricultural instruments. Apply through the Society giving age, experience and salary expected.

0196 Shop superintendent, to take charge of finishing department of Pennsylvania concern manufacturing bearing bronze; will be expected to take charge of the power plant and all machinery in the works. Want live, progressive man who can secure the maximum output and work with management in the endeavor to manufacture quickly and cheaply.

0197 Two sales engineers on general line of machinery for New England territory. New York concern.

0198 Superintendent for manufacturing company. Location Michigan. Metal working plant, principally in brass and copper, employing about 1000 men. Prefer experienced and technically educated man; will consider one from any line of manufacturing or railroad work who can show record for efficient handling of labor and economy in operation.

0199 Professor of mechanical engineering for Imperial Pei-Yang University, Tientsin, China; one with several years' experience in teaching and practical work, from 28 to 30 years of age or older. Apply through the Society not later than October 15.

0200 A large manufacturing and engineering company in New York requires a young engineer in the proposition and estimating department, where the opportunities for advancement are exceptionally good. A technical graduate of 1911 or 1912 preferred.

0201 Ohio concern want an engineer, preferably college graduate, with several years' experience in power plant work, particularly in the following up of economies in operation of boilers, and if possible experience with miscellaneous uses of steam in manufacturing processes.

0202 Young engineering graduate for Ohio concern. Position would be with view to acquiring experience and working up to more important work in engineering department.

0203 Engineering salesman with selling record as well as compressed air and hydraulic experience. Location, New York. Apply through the Society.

0204 Assistant to chief engineer in boiler shop department of Pennsylvania concern. Young technical graduate preferred, who after a few months' experience would be competent to get out bills of material for riveted and welded work and able to check drawings for same.

0205 Draftsman on ammonia and brine piping, thoroughly experienced, required by large ice-machine company. State experience, age and salary expected. Apply through the Society.

0206 Wanted: a mechanical engineer with experience in the design of steam and electrical machinery; one somewhat familiar with power plant machinery, who is competent to fill the position of chief engineer in large manufacturing concern. Location eastern Pennsylvania. Apply through the Society.

0207 Mechanical draftsman experienced in general valve construction, or one acquainted with the detail make-up of sanitary plumbing fixtures. A permanent position and good salary to the right party. State salary and experience. Apply through the Society. Location New York.

0208 General superintendent of company in Middle West, manufacturing large engines and special machinery, employing 300 men. Experience required in drafting, machine shop and foundry. Apply through the Society.

0209 Draftsman for position in Morenci, Arizona. Salary $125 a month with opportunity for advancement. One with experience in concentrator work preferred.

MEN AVAILABLE

497 Associate, aged 35, desires position of some responsibility in Philadelphia or suburbs. Sixteen years' experience in civil, structural and mechanical lines, mainly the last, as draftsman, squad foreman, chief draftsman and checker at rolling, steel and pipe mills, by-product coke ovens and on chemical apparatus, steel cars and locomotives.

498 Mechanical and electrical engineer, Cornell graduate, 12 years' experience in design, construction and operation of steam, hydraulic and gas plants for public service and industrial companies, building construction, factory superintendence and appraisals of public service companies, desires permanent position with manufacturing company, consulting engineers or operating company in or near New York City.

499 Member, now general superintendent of machine tool building shop, also experienced in manufacturing electric machinery, engines, and automobiles, desires change. Up-to-date in shop management and methods, machinery and tools. Age 37, married, technical education. Salary $3000.

500 Mechanical engineer, superintendent, chief draftsman or production manager, with practical shop, technical and commercial experience, expert in

the up-to-date manufacture of interchangeable parts, resourceful in design and process for reducing costs and increasing production, good organizer and system man with excellent executive ability, A-1 references, desires permanent position.

501 Junior, technical graduate, aged 28, four years' practical experience in engineering lines, desires a position as superintendent, or assistant superintendent with manufacturing concern.

502 Technical graduate, 24 years of age, at present employed as assistant to superintendent in large water gas plant, desires to make a change to concern having an opening along mechanical or sales engineering lines. Opportunity and scope desired.

503 Engineer, married, 12 years' practical experience in the manufacture of typewriters, adding machines, automobiles, etc., specialist in the design of all kinds of automatic machinery and development of rapid interchangeable manufacturing equipment. Salary not less than $2400.

504 Junior, technical graduate, mechanical engineer, age 30, experience in locomotive shop on combined erecting and repair work and in steel plant; three years' experience in heavy, medium and light machinery including small tool factory, desires position as assistant to a progressive superintendent.

505 Graduate engineer desires position as mechanical superintendent, master mechanic or engineer of construction in power plant work.

506 Mechanical engineer desires position as purchasing agent or business connection with manufacturing firm. Age 28, college training, five years' experience as shop foreman, sales engineer, and contractor. At present with large locomotive works.

507 Technical graduate in mechanical engineering, 25 years of age, 3 years' experience in machine shop and engineering department of large engine manufacturing plant. Desires to become connected with a concern having an opening along mechanical lines.

508 Member, technical graduate, desires position as sales engineer. Experienced as skilled mechanic, chief draftsman and superintendent of construction of power plant equipment. Four years in New York and one in Chicago as sales engineer.

509 Member, with technical education and training and subsequent managerial experience in commercial lines, is open for association and investment, if necessary, with one or more engineers who may have a broad basic patent or device for promotion, development and manufacture.

510 Member, now employed, desires change. Several years' experience as mechanical engineer and consulting engineer for large corporations. Age 54, married, first class references. No objection to location so long as climate conditions will not impair health. Salary $3000 to $4000.

511 Member, at present employed, desires to improve position. Eighteen years' experience especially in internal-combustion engine lines. Qualified for superintendent, assistant manager, designing engineer, or salesman for gas or oil engine concern.

512 Junior, mechanical and electrical engineer, technical graduate, 30 years old, single, experienced in hydro-electric developments, including hydrograph work, design of transmission lines, towers, sub-stations, etc., with operation of same, desires position as chief engineer or designing engineer with large firm of contracting engineers. Best references as to habits and ability. At present employed.

513 Member, hydraulic and electrical engineer, M. I. T. graduate, manufacturing, consulting and operating experience, 42 years old; recently connected with machinery for some of the largest and typical hydraulic projects.

514 Electrical and mechanical engineer open for engagement, Philadelphia or vicinity preferred. Unusual technical training and ability. Experienced in organization of engineering work, research, development of new materials and processes, design of automatic machinery, layout and installation of new factory departments and engineering supervision of manufacturing processes. Exceptional credentials as to character and ability. For further information, address P. O. Box No. 4040, West Philadelphia, Pa.

515 Position as superintendent or other executive position in manufacturing department; 12 years' successful experience in charge of the manufacture of harvesting machines and agricultural implements; 36 years old, married. Best references relative to ability and character.

516 Superintendent or mechanical engineer, Associate member, with 16 years of shop and office experience in design and manufacture of special machinery, soil pipe and fittings, valves, plumbers and steam fitters supplies with proven ability, desires to connect with concern where energy, efficiency, and mechanical knowledge is required of all departments from foundry to shipping. At present employed.

517 Technical graduate, age 30, married, four years as assistant engineer in the manufacture of contractors and railroad equipment, four years engineer on power and industrial plants, railroad car shops and terminals, inspecting, designing, estimating costs and construction, selling. At present employed. Will consider improved position with engineers, contractors, or manufacturers. Personal interview in Philadelphia or New York.

ACCESSIONS TO THE LIBRARY

WITH COMMENTS BY THE LIBRARIAN

This list includes only accessions to the library of this Society. Lists of accessions to the libraries of the A. I. E. E. and A. I. M. E. can be secured on request from Calvin W. Rice, Secretary, Am. Soc. M. E.

AMERICAN ASSOCIATION OF DEMURRAGE OFFICERS. Proc., 23d Annual Convention. *San Francisco, 1912.* Gift of the association.

DAS AUTOGENE SCHWEISSEN UND SCHNEIDEN MIT SAUERSTOFF, Felix Kagerer. *Wien, 1909.*

BETON UND EISEN. I. Erganzungsheft zum Jahrgang 1912. *Berlin, 1912.*

DIE DAMPFTURBINE ALS SCHIFFSMOTOR, Karl Besig. *Berlin, 1911.*

ELEMENTS OF DRAWING, George F. Blessing and Lewis A. Darling. *New York, J. Wiley & Sons, 1912.*

The work is based on the drawing room courses required of first year students in mechanical engineering in Cornell University. It is intended rather as a reference book than a text book, each drawing plate having certain paragraphs assigned. It includes a somewhat full treatment of isometric drawing.

DIE ESSIG FABRIKATION, Josef Bersch. *Wien, 1907.*

HUMAN FACTOR IN WORKS MANAGEMENT, James Hartness. *New York, McGraw-Hill Book Co., 1912.*

The title gives a hint as to the subject matter of the work, which deserves careful perusal, being the first attempt to apply philosophical principles to a discussion of the mental habit of the workman and works manager.

KRANE. Textband u. Tafelband. Anton Böttcher. *Munchen, 1906.*

DER MODERNE SCHIFFBAU, G. Neudeck, B. Schulz and R. Blochmann. vols. 1-2. *Leipzig, 1912.*

NEW ORLEANS SEWERAGE AND WATER BOARD. 24th Semi-Annual Report. *New Orleans, 1911.* Gift of the board.

NEW YORK CITY BOARD OF WATER SUPPLY. Contract 111 for the Construction of Three Concrete Bridges and Approaches in the Towns of Olive and Marbletown, Ulster Co., N. Y., 1912. Gift of the board.

ONTARIO POWER COMPANY OF NIAGARA FALLS. Maps of Power Districts. Gift of Niagara, Lockport and Ontario Power Co.

PACIFIC GAS AND ELECTRIC MAGAZINE. vol. 3, 1911–1912. *San Francisco, 1911–1912.* Gift of the magazine.

PENNSYLVANIA AND ITS MANIFOLD ACTIVITIES. Prepared by G. C. Whidden and W. H. Schoff for the 12th International Congress of Navigation. *Philadelphia, 1912.*

Poor's Manual of Industrials, 1912. *New York, 1912.*

Port and City of Philadelphia. Prepared by F. H. Taylor and W. H. Schoff for the 12th International Congress of Navigation. *Philadelphia, 1912.*

Die Statik des Kranbaues, W. L. Andrée. *München, 1908.*

Syllabus of Mathematics. *Ithaca, 1912.* Gift of Society for the Promotion of Engineering Education.

Gift of Z. A. Willard

Final Report on the Smoke Nuisance. *February 26, 1912.*

Report on the Smoke Nuisance. *November 19, 1907.*

Smoke Nuisance and Health.

Smoke Nuisance in Europe and America; its Menace and its Cure.

UNITED ENGINEERING SOCIETY

L. L. Poates and Co's. Complete Atlas of the World. *New York, 1912.*

"Titanic" Disaster. Hearings before a Sub-committee of the Committee on Commerce, United States Senate, 62d Congress, Second Session. *Washington, 1912.* Gift of U. S. Senate.

Trow's General Directory of the Boroughs of Manhattan and Bronx, City of New York, vol. 126. *New York, 1912.*

Gift of Association of American Steel Manufacturers

Standard Specifications for Concrete Reinforcement Bars Rolled from Billets, as adopted, 1910. Revised 1912.

Standard Specifications for Structural and Boiler Steel as adopted August 9, 1895. Revised 1896, 1902, 1903 and 1912.

Standard Specifications for Rail Steel Concrete Reinforcement Bars, as adopted 1912.

EXCHANGES

American Society of Refrigerating Engineers. Trans., vol. 5. *New York, 1909.*

Brooklyn Engineers Club. Proc., 1911. *Brooklyn, 1912.*

International Association for Testing Materials. Proc. 6th Congress of the International Association, September 1912.

Iron and Steel Institute. Carnegie Scholarship Memoirs. vol. 4. *London, 1912.*

——Charter, By-Laws and List of Members, July 1912. *London, 1912.*

Wisconsin Gas Association. Journal of Proc., 8th, 10th and 11th Annual Convention. *Milwaukee, 1909–1911.*

TRADE CATALOGUES

American Blower Co., *Detroit, Mich.* Mechanical draft, 64 pp.

Aquabar Co., *Philadelphia, Pa.* Aquabar waterproofing compound for concrete or cement motors, 56 pp.

Fulton Machine & Vial Co., *Locoville, N. Y.* Vises and pumps, 40 pp.

Hess-Bright Mfg. Co., *Philadelphia, Pa.* Thrust collar bearings; driving clutch thrust bearings; ball bearings in horse vehicles; ball-bearing mountings for vertical armature shaft of electric motor, 5 pp.

Holmes Metallic Packing Co., *Wilkes-Barre, Pa.* Metallic packings, 40 pp.

Illinois Stoker Co., *Alton, Ill.* Chain grate stokers, 31 pp.

Ingersoll-Rand Co., *New York.* Water lifted by compressed air, 67 pp.

International Gas Engine Co., *Cudahy, Wis.* "Ingeco" gas, gasolene, oil and producer gas engines, 64 pp.

M. W. Kellogg Co., *New York.* Smoke stacks, 48 pp.; welding pipe flanges and nozzles, 12 pp.

Metallic Packing & Mfg. Co., *Elyria, Ohio.* Martell packings, 16 pp.

Newton Machine Tool Works, *Philadelphia, Pa.* Cat. no. 47 of machine tools, 94 pp.

Sowers Mfg. Co., *Buffalo, N. Y.* Boilers, machines, iron castings, 93 pp.

Tight Joint Co., *New York.* Price list of fittings, flanges and valves, 60 pp.

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University of Kansas	Mar. 9, 1909	P. F. Walker	V. H. Hilford	L. L. Browne
New York University	Nov. 9, 1909	C. E. Houghton	Harry Anderson	AndrewHamilton
Univ. of Illinois	Nov. 9, 1909	W. F. M. Goss	L. G. Smith	C. A. Schoessel
Penna. State College	Nov. 9, 1909	J. P. Jackson	J. A. Kinney	H. S. Rogers
Columbia University	Nov. 9, 1909	Chas. E. Lucke	E. W. Stone	E. A. Jarecki
Mass. Inst. of Tech.	Nov. 9, 1909	E. F. Miller	J. G. Russell	J. B. Farwell
Univ. of Cincinnati	Nov. 9, 1909	J. T. Faig	C. W. Lytle	A. O. Hurxthal
Univ. of Wisconsin	Nov. 9, 1909	A. G. Christie	W. D. Moyer	H. C. Prochazka
Univ. of Missouri	Dec. 7, 1909	H. Wade Hibbard	A. C. Edwards	P. A. Tanner
Univ. of Nebraska	Dec. 7, 1909	J. D. Hoffman	P. S. Toney	M. C. Evans
Univ. of Maine	Feb. 8, 1910	Arthur C. Jewett	A. H. Blaisdell	W. B. Emerson
Univ. of Arkansas	Apr.12, 1910	B. N. Wilson	J. A. Dickinson	W. B. Gardner
Yale University	Oct.11, 1910	L. P. Breckenridge	C. E. Booth	O. D. Covell
Rensselaer Poly. Inst.	Dec. 9, 1910	A. M. Greene, Jr.	W. D. Small	O. A. Van Denburgh, Jr.
State Univ. of Ky.	Jan.10, 1911	F. P. Anderson	J. W. Cary	J. T. Lowe
Ohio State University	Jan.10, 1911	E. A. Hitchcock	J. P. Stewart	H. F. Belt
Washington University	Mar.10,1911	E. L. Ohle	E. Dougherty	E. L. Lacey
Lehigh University	June 2, 1911	H. A. S. Howarth	E. E. Finn	Nevin H. Guth
Univ. of California	Feb.13, 1912	Joseph N. Le Conte	G. M. Simonson	G. H. Hagar

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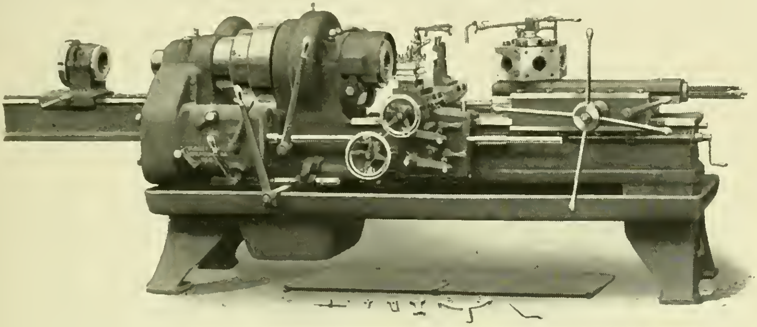

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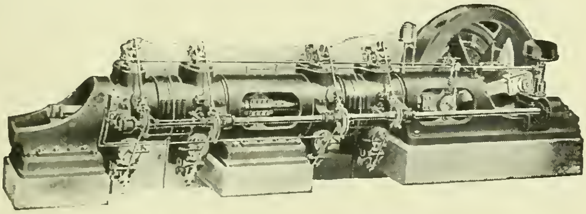

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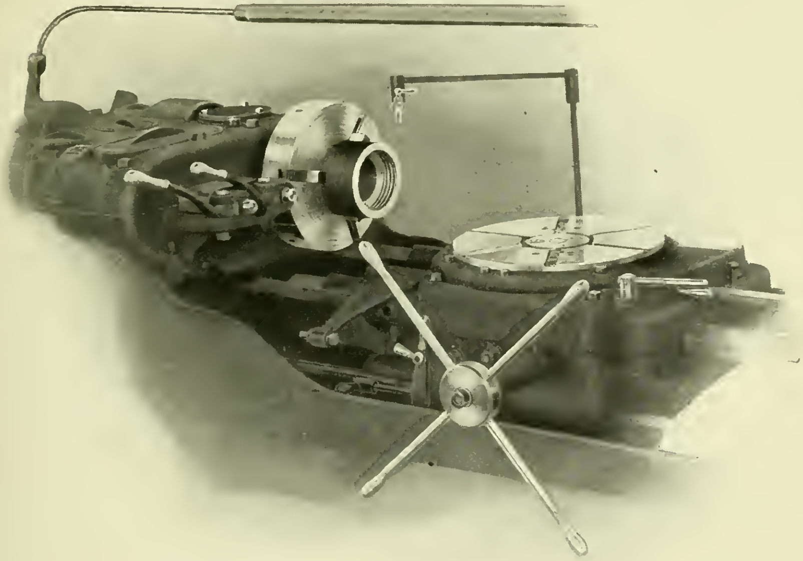

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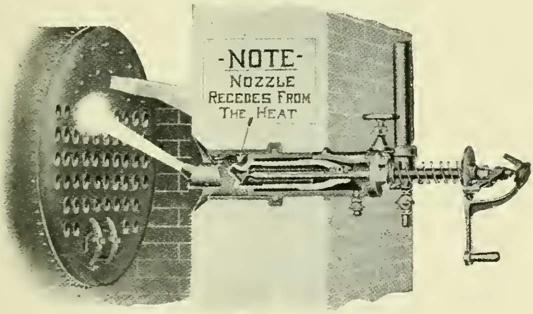

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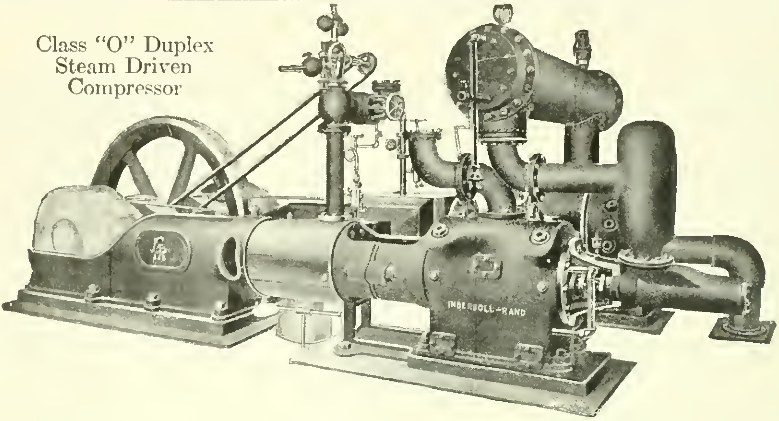

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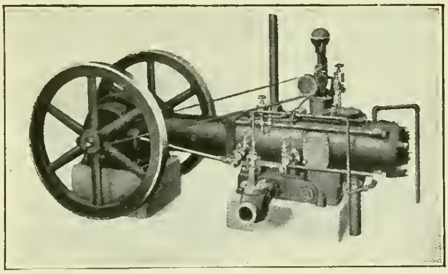

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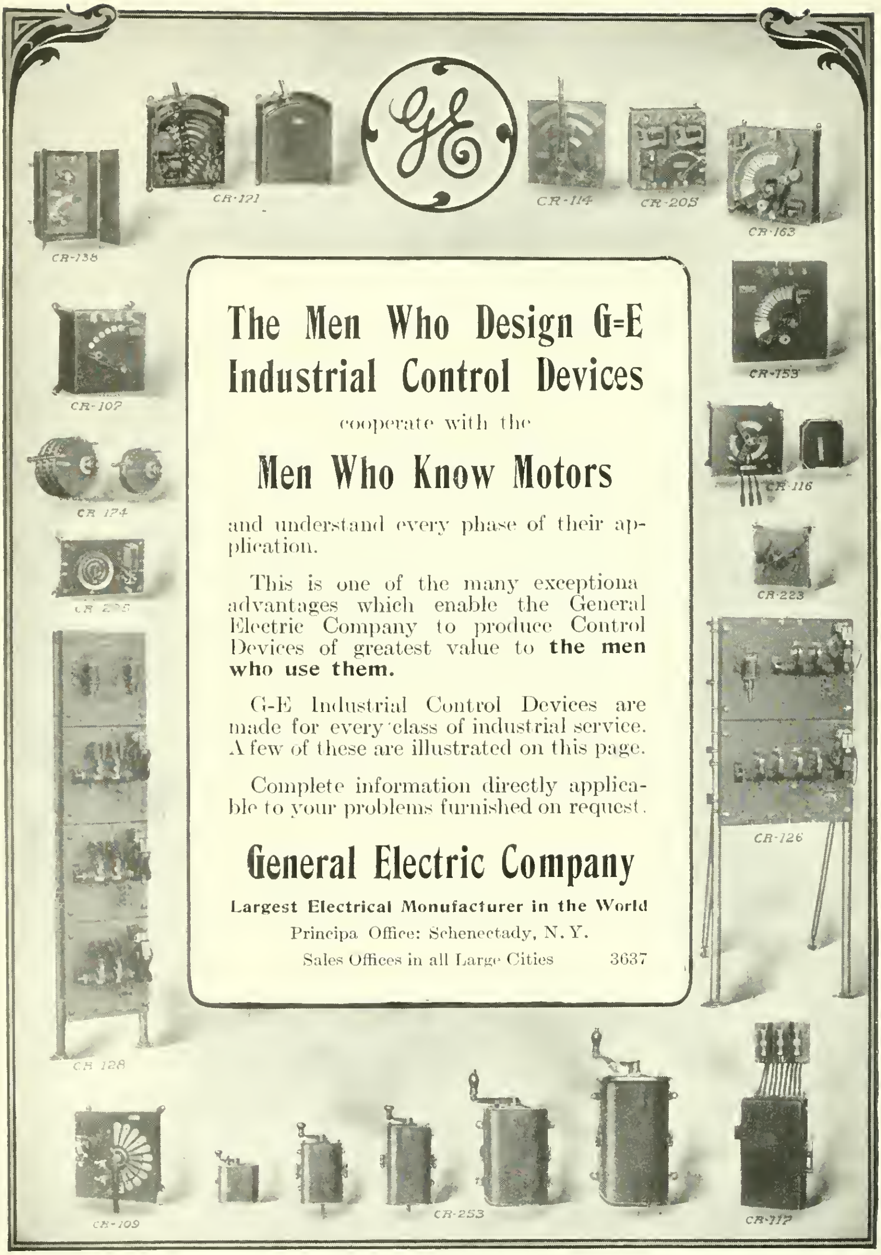

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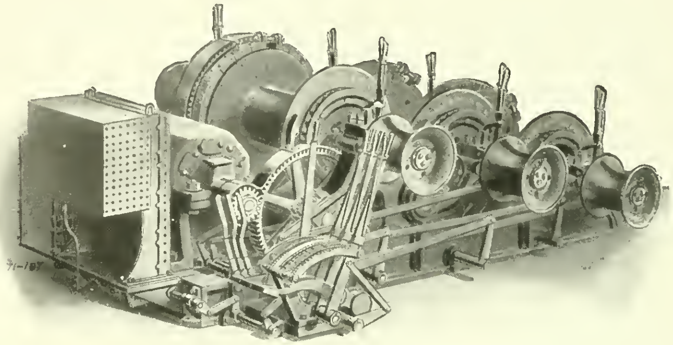

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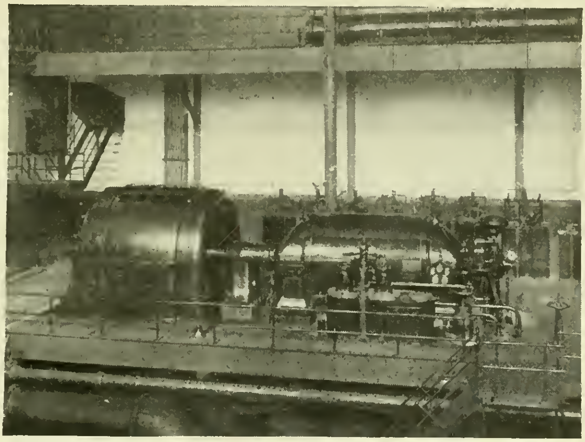

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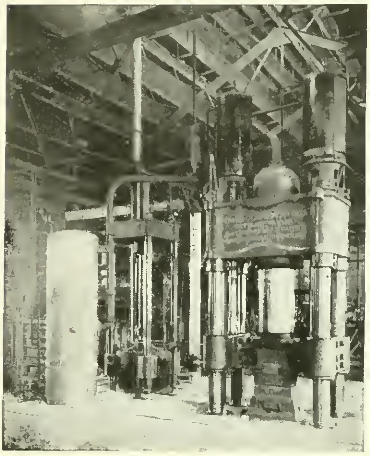

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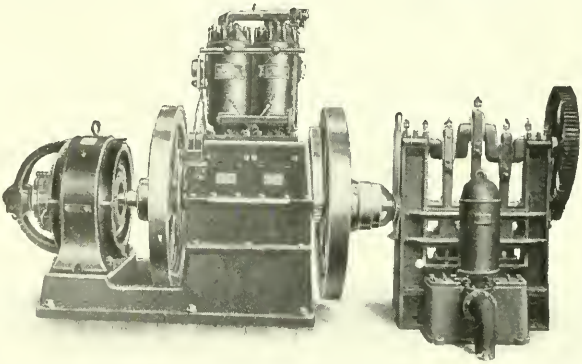

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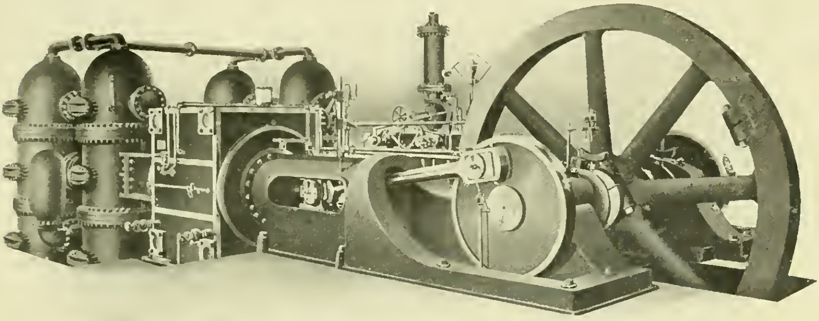

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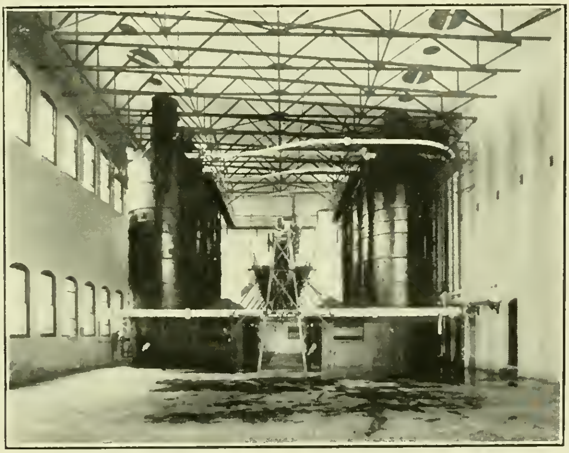

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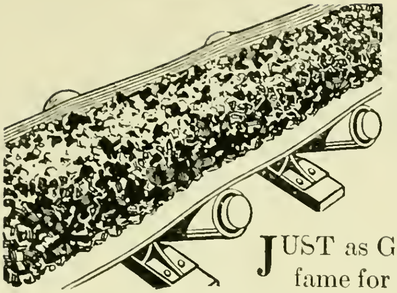

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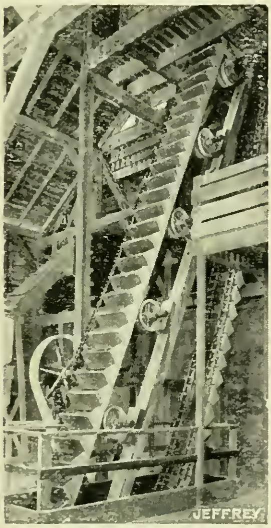

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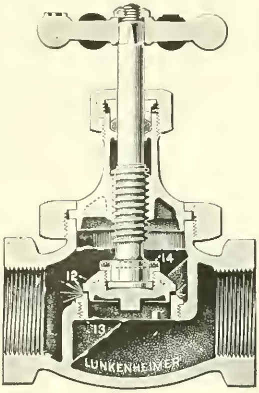

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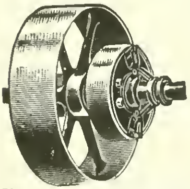

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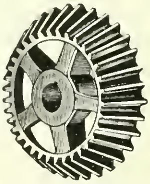

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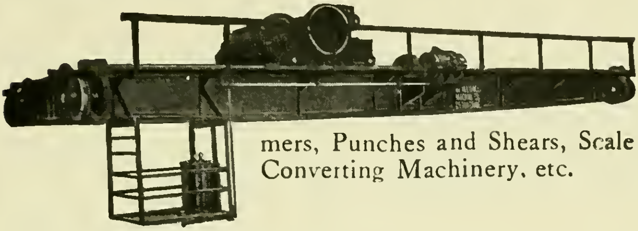

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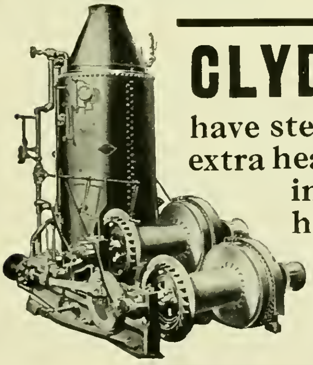

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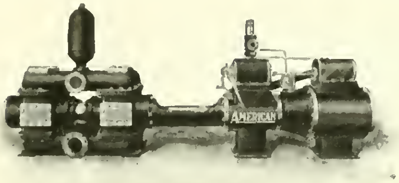

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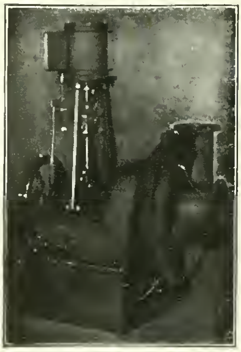

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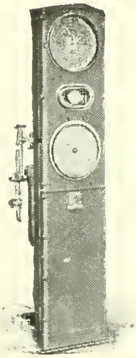

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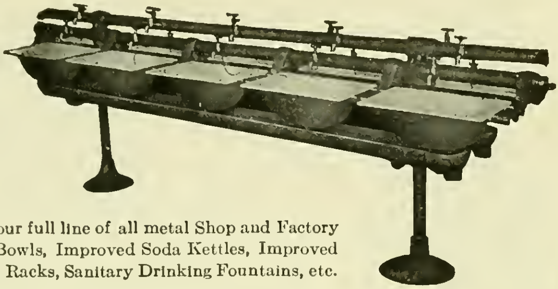

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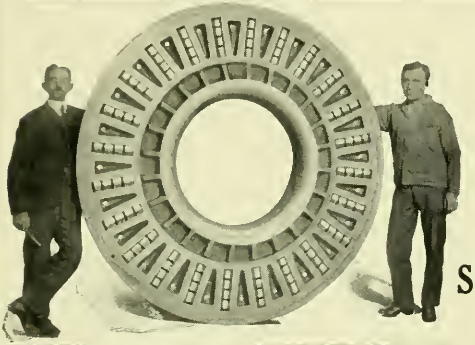

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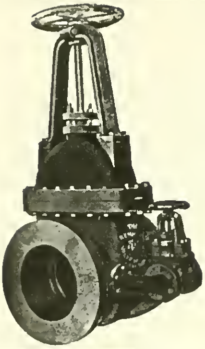

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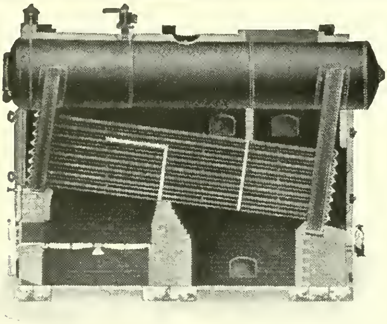

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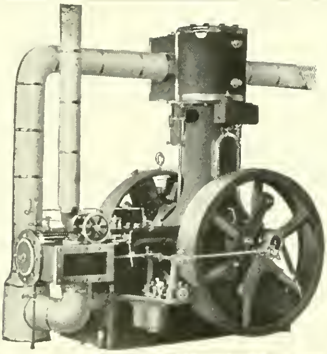

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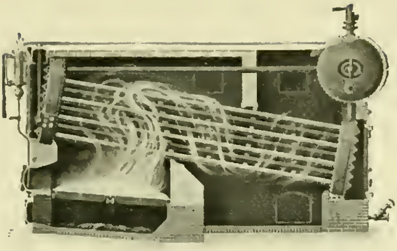

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THE AMERICAN SOCIETY OF
MECHANICAL ENGINEERS

29 WEST 39th STREET, NEW YORK

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NORTHEASTERN UNIVERSITY
ARY

THE JOURNAL
of
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

NOVEMBER 1912

35 CENTS A COPY $3 A YEAR

ANNUAL MEETING: NEW YORK, DECEMBER 3-6
MONTHLY MEETINGS: NEW YORK, NOVEMBER 12; NEW HAVEN, NOVEMBER 13; BOSTON, NOVEMBER 15

INVITATIONS TO THE ANNUAL MEETING

The program of the Annual Meeting of the Society, to be held in New York, December 3-6, will be the most comprehensive ever offered. A number of special sessions will be held which will doubtless attract widespread interest owing to their unusual character and value. One entire day will be devoted to Economic Administration, in which all executives will be interested and the Society will be pleased to issue invitations for that session.

The Society will be pleased to send upon the request of any member, an invitation to the meeting to persons interested. It is hoped that no one will miss this opportunity to bring the work of the Society to the attention of every engineer of prominence.

A brief outline of the program will be found on page 4.

COMMITTEE ON INCREASE OF MEMBERSHIP

I. E. Moultrop, *Chmn.*
C. W. Aiken
J. V. V. Colwell
R. M. Dixon
W. T. Donnelly
J. P. Illsley
E. B. Katte
H. S. Wynkoop

NOVEMBER 1912 VOL. 34 No. 11

THE JOURNAL

OF

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The Journal is published monthly by The American Society of Mechanical Engineers.
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COMING MEETINGS OF THE SOCIETY

November 12, *New York City,* Engineering Societies Building, 8.15 p. m. Paper: Measuring Efficiency in Manufacturing, Edward B. Passano. Informal dinner, 6.15 p.m. Meet in Society's rooms.

November 13, *New Haven, Conn.*, Mason Laboratory, Yale University, afternoon and evening sessions, 3 p.m. and 8 p.m. Dinner at 6 p.m. at Yale Dining Club. Topic for afternoon session: The Transportation of Materials in Manufacturing Plants.

November 15, *Boston, Mass.* Paper: Rot in Timber used in Slow-Burning Construction, F. J. Hoxie.

December 3-6, *New York City,* Annual Meeting, Engineering Societies Building. See page 4 for tentative program.

THE JOURNAL

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOL 34 NOVEMBER 1912 NUMBER 11

THE ANNUAL MEETING

The Annual Meeting this year will comprise sessions upon a greater variety of topics than have ever been arranged for a meeting. No less than ten sessions are contemplated with papers grouped to form symposiums upon various subjects. This broadening of the Society's work is the result of the sub-committees lately appointed under the direction of the Committee on Meetings. The publication of the papers for the Annual Meeting began in the September number of The Journal, and others will appear in the December number or be printed in pamphlet form for distribution at the meeting and published in later issues of The Journal. The papers already published are as follows:

The Principles of Valuing Property, Henry K. Rowell
Increase of Bore of High-Speed Wheels by Centrifugal Stresses, Sanford A. Moss
Investigation of Efficiency of Worm Gearing for Automobile Transmission, Wm. H. Kenerson
Measurement of Air in Fan Work, Charles H. Treat
The Baltimore Sewerage Pump Valve, A. F. Nagle
The Vauclain Drill, A. C. Vauclain and Henry V. Wille
Tests of a 1000-H.P., 24-Tubes High B. & W. Boiler, B. N. Bump
The V-Notch Weir Method of Measurement, D. Robert Yarnall
Dimensions of Boiler Chimneys for Crude Oil, C. R. Weymouth
Air in Surface Condensation, George A. Orrok
The Centrifugal Blower for High Pressures, Henry F. Schmidt
Measurement of Natural Gas, Thos. R. Weymouth

One whole session will be devoted to reports of technical committees which it is expected will be of as great interest as any other single feature of the professional sessions. One of the reports is an exhaustive document prepared by the Power Tests Committee. When finally revised and submitted to the Council, this will supersede the earlier reports upon methods for conducting trials of pumping engines, locomotives, steam boilers and steam engines, besides taking up the testing of other apparatus, such as gas and oil engines, waterwheels, and compressors, blowers and fans. The latest Society report upon any of these subjects was in 1902 and the earliest in 1890, twenty-two years ago, so that the need for revision is evident.

The general arrangement of the sessions will be substantially as follows:

TENTATIVE PROGRAM

Tuesday Evening, December 3

President's Address followed by the reception to the president-elect, the retiring president, Dr. Alex. C. Humphreys, and their ladies.

Wednesday Morning, December 4

Business meeting, followed by simultaneous sessions, under the direction of the Gas Power Section and the sub-committees on Machine Shop Practice and Textiles.

Wednesday Afternoon

Reports of Committees, Power Tests, Symbols, etc.

Wednesday Evening

Dinner to Professor John E. Sweet, Honorary Member and Past President, in celebration of his eightieth birthday and in recognition of his services to the engineering profession, given by his friends and members of the Society.

Thursday Morning, December 5

Simultaneous sessions under the direction of the sub-committees on Railroads, Iron and Steel, and Cement Manufacture.

Thursday Afternoon

Sessions on the Power Plant, and on Hydraulic and Pneumatic Apparatus, including centrifugal pumps, blowers and the measurement of flow of fluids.

Thursday Evening

Reunion and dance, with collation.

Friday Morning, December 6

Session of the Sub-Committee on Administration, presenting a review of the present state of the art of management.

The list of papers for the Machine Shop session is a strong one and includes the results of tests and investigations as well as machine shop operations. At the Textile session the valuation of property and the special requirements of power plants for textile mills are the subjects for discussion. The sub-committee on Railroads have papers on train lighting, electric locomotives and the locomotive problem, and at the Iron and Steel session the most advanced achievements in iron and steel manufacture will be the subject. At the Cement session the subjects of depreciation and obsolescence and of accidents are to be treated.

On Friday the whole session, and the whole day if necessary, will be devoted to a review of the present state of the art of industrial management.

RAILROAD TRANSPORTATION

Arrangements for hotel, transportation and Pullman car accommodations should be made personally.

For members and guests attending the Annual Meeting in New York, December 3–6, 1912, the special rate of a fare and three-fifths for the round trip, on the certificate plan, is granted when the regular fare is 75 cents and upwards, from territory specified below.

a Buy your ticket at full fare for the going journey, between November 29 and December 5 inclusive, and get a certificate, *not a receipt*, securing these at least half an hour before the departure of the train.

b Certificates are not kept at all stations. If your station agent has not certificates and through tickets, he will tell you the nearest station where they can be obtained. Buy a local ticket to that point and there get your certificate and through ticket.

c On arrival at the meeting, present your certificate to the registration desk at the Headquarters. A fee of 25 cents will be collected for each certificate validated. No certificate can be validated after December 6.

d An agent of the Trunk Line Association will validate certificates, December 4, 5, 6. No refund of fare will be made on account of failure to have certificate validated.

e One-hundred certificates and round trip tickets must be presented for validation before the plan is operative. This makes it important to show the return portion of your round trip ticket at Headquarters.

f If certificate is validated, a return ticket to destination can be purchased, up to December 10, on the same route over which the purchaser came, at three-fifths the rate.

This special rate is granted only for the Trunk Line Association, including:

All of New York east of a line running from Buffalo to Salamanca, all of Pennsylvania east of the Ohio River, all of New Jersey, Delaware and Maryland; also that portion of West Virginia and Virginia north of a line running through Huntington, Charleston, White Sulphur Springs, Charlottesville, and Washington, D. C.

THE JOINT MEETING IN LEIPZIG

General plans for the itinerary of the Society's visit to the German Verein have been presented to the Committee on Arrangements by Dr. Matschoss, their special representative, and their adoption now depends mainly on an approximately correct list of members and guests who will compose the party.

Preliminary estimates of the cost of the excursion here given will, it is hoped, enable members to reply more definitely to our circular of August 27, for the cost of steamship and railroad fares depends directly on the number of participants.

We can accommodate from 160 to 200 first-class passengers; about 80 on two promenade decks, the others on the upper and main decks. Prices, according to location, range from $90 to $130 per berth. This includes first-class passage from New York to the German port, with meals and hot and cold salt water baths, and railroad fare to Bremen or Hamburg. The railroad fare from there to Leipzig will be $5 each on regular express trains, but if at least 160 members and guests take part, a special train can be furnished at about $2.50 per person. The excursion from Leipzig to Dresden, Berlin, Düsseldorf, Essen, Cologne, Wiesbaden, Frankfurt, Heidelberg, Nürnberg and Munich will cost from $23 to $26 each, with probably a similar liberal deduction if a special train is chartered for not less than 160 passengers.

German railways have first, second, third and fourth-class carriages. There is no difference between first and second-class except the color of the upholstery and the seating capacity, first-class having six seats and second-class eight seats per section. As a rule, well-to-do Germans travel in second-class carriages, first-class being patronized by high officials and foreigners unfamiliar with the customs of the country. For short trips of less than five hours, most

Germans on their summer vacation tours travel third-class, the carriages of which are roomy and clean, with plain wooden seats.

The figures above given are for second-class coaches throughout. Dining cars accommodate first and second-class passengers only. Hotel accommodations can be had at American prices if American style and luxury are desired. But the hotels patronized by native travelers are much cheaper, serve excellent meals and are scrupulously neat. Single rooms with private bath cost $2.50 to $3.50 per day; double rooms from $3.50 to $6.00. Single rooms without bath from $1.00 to $2.00; double rooms from $2.50 to $5.00, and excellent baths can be had on the same floor for 35 to 50 cents. In every city there are first-class swimming baths.

Good meals on dining cars and excellent meals in hotels range from 35 cents for breakfast to 75 cents and one dollar for dinner. Germans generally dine at midday, and take coffee or tea in public gardens where a band plays, and supper in garden restaurants; rates about 25 per cent less than in the United States.

In the matter of meals there is a radical difference between hotels in Europe and America. The steam table which reduces all viands to mediocre similarity is unknown in Europe. Meals are served at stated hours fresh from the chef's hands, each course distinguishable without reference to the bill of fare, incidentally reducing the cost of service. The result is that meals in the smaller and cheaper hotels are much better than in houses of the same class here.

With the very reliable Baedeker in your pocket you can travel on the continent at much less cost than here.

A gentleman who recently traveled in Germany and France with two ladies for three and one-half months reports total expense, exclusive of steamer fare, at $5.10 per day per person.

Committee on Arrangements

AMENDMENT TO BY-LAW 18

At a meeting of the Council, October 8, By-Law 18 was amended to read as follows:

B 18 The Council at any meeting may, in its discretion, permanently remit the dues of any Member of the Society who has paid thirty years' dues or who shall have reached the age of seventy years after having paid twenty-five years' dues, provided that notice of such proposed action shall have concurred in recommending that this action be taken. The Council may, in its discretion, restore to membership any person dropped from the rolls for non-payment of dues, or otherwise, upon such terms and conditions as it may at the time deem best for the interests of the Society.

MEETINGS OF STUDENT BRANCHES

With the reopening of the universities, meetings of the Student Branches of the Society, which now number twenty-five with a total membership of 700, have been resumed with customary vigor. Although it is possible to publish no more than very brief accounts of these meetings from time to time, much work of real professional merit is being accomplished in these branches, which deserves wider mention. In some of the branches, outside speakers of prominence in the engineering world are secured to address the college at intervals on vital aspects of the profession, while in others members meet to discuss the papers presented at meetings of the Society. Original papers are also secured and read at many of the branch meetings, and at least one of the colleges held last year a small convention, with afternoon and evening sessions and a carefully arranged program.

The Armour Student Branch opened its meetings for the season on September 19, with its annual smoker at which Prof. G. F. Gebhardt, Prof. A. J. Frith and Prof. C. E. Paul, Members of the Society, were the speakers. On October 9 a joint meeting was held with the student branch of the American Institute of Electrical Engineers, C. K. Baldwin, Mem.Am.Soc.M.E., giving an illustrated lecture on Belt Conveyors.

At the first meeting of the Ohio State University student branch, Prof. Wm. T. Magruder, Mem.Am.Soc.M.E., gave an interesting talk on the Society, its growth and what it stands for, with special reference to the value of affiliation with it.

A well-attended meeting was held at the Polytechnic Institute of Brooklyn on the evening of October 5, for the purpose of laying out plans for the coming season, for which the prospects are excellent. B. L. Heustis was elected president for the year, and A. Bielek, secretary.

The Student Branch of the State University of Kentucky elected the following officers at a recent meeting: R. R. Taliaferro, chairman, R. E. Mattingly, vice-chairman, F. J. Forsyth, secretary, and D. W. Perry, treasurer.

At the University of California, J. F. Ball was elected to the office of chairman at a meeting on September 14, and Jesse Blair to the office of vice-chairman.

The University of Missouri Student Branch held a meeting on September 30, at which Prof. H. W. Hibbard, Mem.Am.Soc.M.E., gave a talk on the value of affiliation. The following officers were elected: F. I. Kemp, president, R. Runge, secretary-treasurer, J. H. Pound, corresponding secretary.

Meetings were resumed by the University of Nebraska Student Branch on October 1, at which short talks on affiliation were given by Prof. J. D. Hoffman and Professor Raber.

The University of Wisconsin Student Branch held their election of officers on October 17, with the following result: W. A. Moyer, president, A. G. Peter, vice-president, W. K. Fitch, secretary, W. H. Schleck, treasurer. Committees were appointed by the president and the work for the year planned.

At Yale University a meeting of the Student Branch was held on October 11, at which Prof. L. P. Breckenridge, Mem.Am.Soc.M.E., was the principal speaker. Plans for the year were outlined by the president, C. E. Booth, and an interesting demonstration was given of a new combined stereopticon and reflectoscope, just installed at the Sheffield Scientific School.

DR. VON MILLER ELECTED TO HONORARY MEMBERSHIP

At a meeting of the Council, October 8, 1912, Dr. Oscar von Miller, director of the German Museum in Munich, was elected to honorary membership in the Society. Dr. von Miller who, in addition to his connection with the German Museum, is this year president of the Verein deutscher Ingenieure and a civil engineer of the city of Munich, has been prominent for a number of years in his profession. He was the designer of the electric central station in Kassel, the first central station in Germany with a converter from alternating to direct current, and also the central station in Heilbroün, the first alternating-current installation in Germany. He also designed and directed the construction of the electric trunk line Mecklenbeuren-Tettnang and the central stations at Nürnberg, Strassburg, Wiesbaden, Bozen-Meran, Riga, and others. He is the author of a book on the Supply of Electricity in Cities, and has received decorations from the governments of Italy, Sweden, Bavaria, the Grand Duchy of Baden, and from Württemberg.

Dr. von Miller was one of the founders of the German Museum of Masterpieces of Natural Sciences and Technology, of which he is now the director.

NEW SECTIONS OF THE SOCIETY

In addition to the meetings of the Society which are now held in New York, Boston, Philadelphia, St. Louis, San Francisco, and New Haven, committees have been organized in Cincinnati and Chicago in charge of meetings in those cities, with the following personnel: In Cincinnati, A. L. DeLeeuw, chairman, J. T. Faig, secretary, W. G. Franz, G. W. Galbraith, L. H. Thullen; in Chicago, Paul P. Bird, H. A. Bogardus, G. F. Gebhardt and A. L. Rice.

On October 17, the first meeting of the Cincinnati Section was held, jointly with the Engineers Club of that city. A. L. DeLeeuw, chairman of the section, read a paper on The Trend of Modern Engineering, which treated of the gradual change of the scope of the engineer from an inventor to a true economist, who sought to get the greatest amount of product with the least amount of effort or expenditure. The latest phase of this subject was especially considered, namely, scientific management, and how the task and bonus plan in connection with instruction sheets leads to economies, not only by its direct effect on the workingman, but also because it brings to the attention of the management every point of the equipment not up to its highest efficiency. The paper further discussed the effects of this system on the relation between employer and employee and how it establishes a well-defined contract between them, producing a beneficial effect and becoming in itself a source of economy.

DINNER TO DR. MATSCHOSS

At the conclusion of the Council meeting on October 8, an informal dinner was tendered by the members of the Council to Dr. Conrad Matschoss, Dozent of the Technische Hochschule of Berlin, an official representative of the Verein deutscher Ingenieure.

In Dr. Matschoss's remarks he spoke of his success in his tour of the United States in interesting men of affairs and engineers in the writing of their biographies and histories of the technical arts, so that there may be available in the future a record of the development of each of the industries. He remarked that the conditions in particular ages may be obtained from literature, but that no convenient report of the progress made in any art is at present obtainable, with a few exceptions. Dr. Matschoss also spoke of the increasing interest which every young man ought to have in his profession, and the importance of developing this respect and zeal for one's work.

Mr. Swasey, who took the chair after Dr. Humphreys was called away, pledged the company to take up the work as outlined by Dr. Matschoss, and it is the hope of all that in the near future several of our honorary members and others prominent in the engineering profession will make good their promise to write autobiographical notes which the Society may publish uniformly with the edition of the Life of John Fritz, brought out under the auspices of the Society.

The meeting concluded with an expression of appreciation to the Verein deutscher Ingenieure of their invitation to the Society to meet with them in Germany next year and to make a tour of the industrial centers.

REPORT OF COMMITTEE ON PATENT LAWS

The following report on the patent law situation has been made by the committee appointed by the Council to investigate this subject, in which every member is directly or indirectly interested. The committee attended the patent conference held last spring in Washington between the bar associations and the engineering societies of the country, and have also been requested by the Council to continue to keep in touch with the situation and to report to them from time to time regarding recent developments.

To the Council:

Pursuant to our appointment on April 9, 1912, by the Council as a Committee on Patent Laws, we have continued to study the patent situation in Washington and the course of legislation which has been introduced in the House on the subject of Patents, amounting to more than 50 bills, most of which appear to be not in the interest of inventors and patentees.

In the early part of May, the President sent a message to Congress recommending the appointment of a competent mixed commission to make a thorough study of the patent law and patent practice and to report its findings to Congress as a basis of legislation.

Your Committee believes that the recommendations of the President, as embodied in his message, are most wise, that there is great danger in any patent legislation that is not based on carefully compiled information, that the appointment of a commission is the only safe procedure, and that the President's recommendations should have the earnest support of all engineers, patentees and others who are interested in the preservation and improvement of our patent system.

Respectfully submitted,

W. H. Blauvelt
B. F. Wood
Committee on Patent Laws

INDEX TO TRANSACTIONS

It is the expectation of the Publication Committee to make this coming season an index of the individual papers in the Transactions, from the first volume up to the present time. It is hoped to make the index a model of its kind, through the coöperation of the librarian of the United Engineering Societies and of others experienced in work of this character.

BOOKS ON AERIAL ENGINEERING

The library of the United Engineering Societies now possesses an important collection of works on aeronautics to which the attention of the membership is called.

At the head of the list may be placed the works of Prof. S. P. Langley (Experiments in Aerodynamics, Washington, 1891, and Memoir on Mechanical Flight, 1911), the dean of the science of aerial engineering. The next place, following the order of development of ideas and not of publication, belongs to the remarkable work of L. P. Mouillard (Le vol sans battement), written sometime in the late eighties or very early nineties of the last century, and clearly establishing the principles of wing warping which gave the mastery of the air to the Wright brothers. A copy of this work was sent to Mr. Octave Chanute in 1894, but was not published until long after the author died, in 1897.

The following works are devoted to the exposition of the fundamental mechanical principles of heavier-than-air flight: Two volumes of Aerial Flight by F. W. Lanchester (Aerodynamics, 1908, and Aerodonetics, 1909), containing a discussion of the theory of aerodynamic support, resistance of bodies in motion to flight, forms of natural flight path, questions of equilibrium, stability in flight, and phenomenon of soaring; Etude de la stabilité de l'aéroplane, by G. de Bothezat (Paris, 1911), examining the necessary and sufficient conditions of the stability of an aeroplane, and for the first time giving a complete mathematical solution of the complicated, but highly important problem of the conditions of stability; Les lois expérimentales de l'aviation, by Alexandre Sée (Paris, 1911), with its companion volume, ordered but not yet received by the library, said to be one of the most remarkable books on the theory of aviation and treating the whole subject in a novel and interesting manner. To the same class belongs the little book on the Force of the Wind, by Prof. Herbert Chatley, dealing with the subject of wind pressure and stresses in structures due to wind, and giving a brief but clear exposition of the stream line theory, of interest to automobile designers as well as aeronauts. Frederic Houssay, Forme, puissance et stabilité des poissons (Paris, 1912), may also be mentioned here as containing many interesting considerations on the inversion and motion of bodies entirely immersed in a fluid, and although primarily devoted to the investigation of the motion of fish, much of its theory applies also to aeroplanes.

On the margin between pure theory of the art and practical design may be placed Les bases et méthodes d'études aérotechniques, by L. Ventou-Duclaux and Robert, covering both the heavier and lighter-than-air flight, and Victor Lougheed's Vehicles of the Air, the first complete exposition of the whole subject in the English language.

It is the intention of the library to have as complete a collection as possible of the publications of the great European aerodynamic laboratories. So far several publications of the laboratory of Champ-de-Mars, conducted by Mr. G. Eiffel, Honorary Member, Am.Soc. M.E., and of the Laboratorio di Costruzioni Aeronautiche del Battalione Specialisti of the Italian War Office, have been secured, and arrangements have been made to obtain the publications of the Aerodynamic Institute of Riabushinsky at Kuchino, near Moscow, Russia.

On design and construction of heavier-than-air machines the library has the following works: Progress in Flying Machines, by Octave Chanute (New York, 1899), now of historical interest only; The Problem of Flight, by Herbert Chatley (London, 1907), an epitome of the knowledge available on the subject of flight at the time when Santos-Dumont and the Wright brothers were demonstrating that flight is actually possible; Navigating the Air, by the Aero Club of America (New York, 1907); The Art of Aviation, by Robert W. A. Brewer (New York, 1910), containing a large amount of practical information upon the construction of aeroplanes and their motive plant, with a sparing use of mathematics; The Principles of Aeroplane Construction, by Rankin Kennedy (New York, 1911), a concise theory of aeroplane construction and elementary treatment of aeroplane design, with calculations worked out numerically on the two systems in use.

For the lighter-than-air flight the library possesses the following: Espitalier, La Technique du ballon (Paris, 1907); Fr. Walker, Aerial

Navigation (New York, 1902), and A. W. Marshall and Henry Greenly, Flying Machines (London, n. d.).

The library receives the following special periodicals on Aeronautics: Rivista Tecnica d'Aeronautica, Organo dell Aero Club d'Italia; La Technique automobile et aérienne; Zeitschrift für Flugtechnik und Motorluftschiffahrt; Deutsche Luftfahrer Zeitschrift; L'Aérophile; Fly; Aircraft; Aeronautics; and the Aeronautical Journal.

THE PRESENT STATE OF THE ART OF INDUSTRIAL MANAGEMENT

MAJORITY REPORT OF SUB-COMMITTEE ON ADMINISTRATION

During the past few years a number of striking phenomena, in connection with industrial management, must have become evident even to the most superficial observer. The more important are:

a The widespread, popular interest in the subject which had its rise in a statement made before the Interstate Commerce Commission, in a hearing on the matter of proposed advances in freight rates by carriers. An attorney for the shippers stated on November 21, 1910, that it was estimated that by the application of newly discovered principles of management "in the railroad operation of this country an economy of $1,000,000 a day is possible," and further that these principles can be applied with equal success "in every form of business activity." This popular interest is shown by the great number of articles published in the daily papers and popular magazines, mediums that give but scant attention to technical subjects, except of the most striking nature.

b The suddenly intensified interest in the subject on the part of employers and business executives in many lines of activity, shown by lectures, addresses, professional papers and reports presented to their associations.

c The opposition of labor unions to the newer methods of management, shown by statements of labor leaders, in a few instances by strikes, and by an attempt to

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The American Society of Mechanical Engineers, 29 West 39th Street, New York. Subject to revision.

prohibit by law the use of some of these methods in Government shops.

d Governmental recognition of the matter shown by the appointment of a special committee of the House of Representatives to investigate systems of management in Government arsenals and shops, which reported in March 1912; by the appointment of a civilian board by the Secretary of the Navy to investigate management in the navy yards, which reported in July 1911; and by Senate bill *S 6172*, now in committee, which is intended to prohibit time study and the payment of premiums or bonus on Government work.

e The rapidity with which literature on the subject has accumulated. One directory of books on business management lists 500 titles, and states that 75 per cent of them have been written within five years.

f The formation of two societies having as an aim the furtherance of the application of the principles of management.

g The separation of persons interested in the matter into two camps, one of enthusiastic advocates, the other of vigorous opponents of what is called the new element in management.

h The unquestionable proof of the advance that can be made in unskilled work, as shoveling material, and in ancient trades, as bricklaying, by the application of the principles of management. This is the most striking phenomenon of all.

THE PRINCIPLES OF MANUFACTURE

2 Before defining the element in the art of management that has given rise to these phenomena, it is necessary to review briefly the beginnings of modern industry. This gives a historical setting from which the present can be more truly judged.

3 Modern industry is stated by some writers to have begun in 1738 when John Wyatt brought out a spinning machine. Others place the period as between 1750 and 1800, when the power loom and steam engine came into being. It was marked by the development of labor-saving machinery. It was brought about by the change from handicraft to manufacture.

4 Early British economists held that the application of the principle of *division of labor* was the basis of manufacture. From Adam Smith's "Wealth of Nations," 1776, we quote:

> "This great increase of the quantity of work which, in consequence of the division of labor, the same number of people are capable of performing, is owing to three different circumstances; first, to the incréase of dexterity in every particular workman; secondly, to the saving of the time which is commonly lost in passing from one species of work to another; and lastly, to the invention of a great number of machines which facilitate and abridge labor, and enable one man to do the work of many."

5 Charles Babbage, the great British mathematician and mechanician, believed that from the above-quoted statement the most important principle was omitted. This omission he supplied as follows in his "Economy of Machinery and Manufacture," 1832:

> "That the master manufacturer, by dividing the work to be executed into different processes, each requiring different degrees of skill and force, can purchase exactly that precise quantity of both which is necessary for each process; whereas, if the whole work were executed by one workman, that person must possess sufficient skill to perform the most difficult, and sufficient strength to execute the most laborious, of the operations into which the art is divided."

6 It appears, however, that another principle is the basic one in the rise of industry. It is the *transference of skill.* The transference of skill from the inventor or designer to the power-driven mechanism brought about the industrial revolution from handicraft to manufacture. It will be necessary to refer to this principle frequently throughout this report, in showing the meaning and position of management in industry.

7 No better single illustration of the application of this principle can be found than in the invention of the lathe slide rest by Henry Maudsley in 1794. This has been ranked as second only to the steam engine in its influence on machinery building, and thus on industrial development. The simple, easily controlled mechanical movements of the slide rest were substituted for the skilful human control of hand tools. So complete has been this transference of skill that today hand tooling is a vanished art in American machine shops. Very few lathe hands can chase a thread with hand tools, yet all can cut good threads on an engine lathe, thanks to the slide rest. After the traditional skill of a trade, or the special, pe-

culiar skill of a designer or inventor, has been transferred to a machine, an operator with little or no previously acquired skill can learn to handle it and turn off the product.

8 An example of the extent to which this transference of skill is carried today is presented by the shoemaking industry. The United Shoe Machinery Company builds some 400 machines used in shoe manufacture. These are so highly organized that the greater part of shoe-shop operatives are unskilled except in a single readily mastered detail of the work. The skill in shoemaking is now in the mechanical equipment of the shops. This transference is a development of the past 50 years.

9 James Nasmyth, a British engineer, inventor of the steam hammer, has this to say in 1851 of the application of this principle in his own works: "The characteristic feature of our modern mechanical improvements, is the introduction of self-acting tool machinery. What every mechanical workman has now to do, and what every boy can do, is not to work himself, but to superintend the beautiful labor of the machine. The whole class of workmen that depend exclusively on their skill is now done away with."

10 Methods of analyzing and recording operations were early developed. Adam Smith records the divisions of the work of manufacturing pins, listing 11 operations. Charles Babbage gives a table (see Appendix No. 1, Table 1) from a French investigator, showing the number of operations, time for each, cost of each, and expense of tools and material for making pins in France in 1760. He gives a similar table for English manufacture in his day (see Appendix No. 1, Table 2, from Economy of Machinery and Manufacture, 1832).

11 He further comments on the use of the watch to time operations. We quote from his instructions to one making such observations and using a skeleton form that he recommends: "In filling up the answers which require numbers, some care should be taken: for instance, if the observer stands with his watch in his hand before a person heading a pin, the workman will almost certainly increase his speed, and the estimate will be too large........The number of operations performed in a given time may be frequently ascertained when the workman is quite unconscious that any person is observing him. Thus the sound made by the motion of a loom may enable the observer

to count the number of strokes per minute, even though he is outside the building in which it is contained."

12 M. Coulomb, the noted French physicist (1736-1806), who had great experience in making such observations, cautions those who may repeat his experiments against being deceived by such circumstances. We translate a single quotation: "I pray (says he) those who wish to repeat them (the experiments) if they have not time to measure the results after several days of work, to observe the workmen at various times during the day without their knowing that they are being watched. We cannot be too well warned of the danger of self-deception in computing either the speed or the effective time of work through an observation of a few minutes."

13 Thus we see the application of the principle of *transference of skill* at the basis of the development of the industry, and an early appreciation of the value of the detailed study of operations in making that transference more complete. But the machine was the viewpoint. It was looked upon as the producing unit. Combined and contrasted with this was a lack of knowledge of scientific principles and their sure application. Charles Babbage treats of this forcefully. We quote:

> "There is perhaps no trade or profession existing in which there is so much quackery, so much ignorance of the scientific principles, and of the history of their own art, with respect to its resources and extent, as is to be met with amongst mechanical projectors."

14 In the same vein he emphasizes the need of accurate drawings as if having in mind the poor quality of the work from the average draftsman of his day: "It can never be too strongly impressed upon the minds of those who are devising new machines (says he) that to make the most perfect drawings of every part tends essentially both to success of the trial, and to economy in arriving at the result."

15 He further points out that there is another important factor in successful industry, in addition to machinery. We read that "in order to succeed in a manufacture, it is necessary not merely to possess good machinery, but that the domestic economy of the factory should be most carefully regulated."

16 These quotations foreshadow modern methods of thinking out the work in advance and transferring this thought to the workmen. The subsequent development has had the effect

of advancing still further the division of labor, and beginning the division of thought. The drafting room presents the first example of the trend, in its collection of engineering data, in its prediction of results and the formation of staff organization.

17 But from the period of the last quotation almost to the present there has been no change in the basic principles discovered and applied in industry. There has been nothing but an extension of those already known. The place of greatest advance has been in the drawing room. The art of machine design has been greatly developed. The last half of the last century saw a tremendous increase in inventions, a tremendous furtherance of the application of transference of skill to machines and tools. The skeleton of an industrial organization of this period, one that was too large for a single executive to manage, consisted of a designing department and a production department, each with a head responsible to the manager.

18 The first of these, the one that was the means of embodying skill in the machinery and tools of production, was highly developed and organized. Experiment, research and detailed study were constantly resorted to, to aid in reaching the desired result. The work was highly specialized and the employees highly paid. Not infrequently the manager or chief executive devoted much of his own time to this part of the business.

19 The production department presented a contrasting condition. The workmen were given the tools and machines designed in the drawing room and using their own unaided skill were expected to produce work of the desired quality and quantity. Except in rare instances no effort was made to transfer the skill of the management to the production department and the employees, or to undertake the division of executive thought. Very little consideration was given to the workmen as a producing unit.

FEATURES OF THE CHANGE

20 Within the past 20 or 25 years certain changes have taken place in the attitude of many production managers toward the problems that they face and the forces and means that they control. An increasing amount of attention is being given to the worker. An early evidence was the development of profit-sharing, premium and bonus systems to reward increased effort

and output. There followed welfare work, industrial betterment movements, the adoption of safeguards and regulations to minimize industrial accidents, the substitution of the principle of accident compensation for employers' liability and an improvement in the physical surroundings and conditions of factories. All of these tendencies have been fostered and to a great extent initiated by employers. But even today these are by no means generally adopted.

21 Another tendency, less pronounced in character, has as its object the improvement of the personal relations between employé and employé and between employé and employer. It is an effort to establish the best of factory working conditions in those things not physical in nature, to develop and maintain a shop atmosphere free from all harassing and hindering influences. It is an attempt to make use of the results of experimental psychology, in improving working conditions.

22 But the most important change and one that comprehends the others, is in the mental attitude toward the problems of production. The tendency is toward an attitude of questioning, of research, of careful investigation of everything affecting the problems in hand, of seeking for exact knowledge and then shaping action on the discovered facts. It has developed the use of time study and motion study as instruments for investigation. the planning department as an agency to put into practice the conclusions drawn from the results of research, and methods of wage payment which stimulate coöperation.

23 All of these changes have affected the production department much more than the designing department. The effect is to extend the principle of transference of skill to production, so that it completely embraces every activity in manufacture. The skill of the management is consciously transferred to all of the operations of the factory. This extension is expressed by these phrases: the drawing room is the planning department of design, and the planning department is the drawing room of production.

NATURE OF THE COMMITTEE'S INVESTIGATION

24 To obtain information on present conditions your committee wrote to the recognized experts, to executives of plants in many lines of industry, to students of industrial problems.

and has had many interviews with men in these various fields. The response to our requests has been in the main most generous. We are deeply indebted to the information thus received for a large portion of the following sections of this report. We are glad to take advantage of this opportunity to express our gratitude to all those who have given aid.

25 Throughout the following pages there is a plentiful use of illustrative quotations. Many of these are taken from correspondence resulting from our investigations. Others are from the mass of literature mentioned in Paragraph 1*e*.

26 On some points diametrically opposed views have been expressed. In such cases we have presented both. In no case has credit been given for these views or quotations, as the information was solicited in confidence.

DEFINITION OF THE NEW ELEMENT IN THE ART OF MANAGEMENT

27 Requests for a definition of the new element in the art of management brought forth a difference of opinion as to its existence. The opposed view is given in the following quotations:

> "I am not aware that a *new* element in the art of management has been discovered———."
>
> "There have been no new discoveries in scientific management of industrial institutions. Common-sense men have used common-sense methods always. The term 'scientific management' is a catch-word which assumes that industrial institutions have not been scientifically managed—which is not the case. My experience and the experience of my friends has been that there has been no new element injected into the art of management."
>
> "In the writer's opinion there is very little that is new about it (the art ot management). There is hardly any part of it that has not been practised by managers for the past 100 years. The trouble is there are not enough managers with sufficient initiative to set the system moving properly."
>
> "——— the problem presented is not the adoption of something entirely new; but rather the extension to every detail of our work of something which we have already tried."

28 Turning now to the other side of the question, from a large number of definitions of this new element we select the following as very nearly conveying, taken together, the complete conception as our investigation has disclosed it:

> "The best designation of the new element I believe to be 'scientific management.' This term already has been adopted quite generally and although frequently misused, carries with it the fundamental idea that the manage-

ment of labor is a process requiring thorough analytical treatment and involving scientific as opposed to 'rule of thumb' methods."

"The writer ventures to define the new element briefly, but broadly, as: The critical observation, accurate description, analysis and classification of all industrial and business phenomena of a recurring nature, including all forms of coöperative human effort and the systematic application of the resulting records to secure the most economical and efficient production and regulation of future phenomena."

"Stripped of technicalities the method of the modern efficiency engineer is simply this: First, to analyze and study each piece of work before it is performed; second, to decide how it can be done with a minimum of wasted motion and energy; third, to instruct the workman so that he may do the work in the manner selected as most efficient."

"The Taylor System is not a method of pay, a specific ruling of account books, not the use of high-speed steel. It is simply an honest, intelligent effort to arrive at the absolute control in every department, to let tabulated and unimpeachable fact take the place of individual opinion; to develop 'team play' to its highest possibility."

"As we conceive it, scientific management consists in the conscious application of the laws inherent in the practise of successful managers and the laws of science in general. It has been called management engineering, which seems more fully to cover its general scope than a science."

29 These quotations convey the ideas of a conscious effort to ascertain and study facts and systematically to apply them in instructing the workmen and in controlling every department of industry. Setting these against the underlying principle of the transference of skill we conceive the prominent element in present-day industrial management to be: *The mental attitude that consciously applies the transference of skill to all the activities of industry.*

30 Here emphasis is placed upon the word *all* for, as shown in Pars. 17 and 18, the restricted application of this principle to machines and tools has been highly developed for a long period. But its conscious application in a broad way to the production departments, and particularly to the workmen, we believe has been made during the last quarter century.

RISE OF THIS MENTAL ATTITUDE

31 The rise of this change of attitude in regard to industrial management is shown in the papers on the subject in the Transactions of this Society. These are 16 in number and are listed in Appendix No. 2 of this report. The period covered is from 1886 to 1908. The practice upon which several were based

extended over a number of years before the paper was presented. Papers on accounting have been excluded.

32 The first, No. 207, classifies *management of works* as a modern art having a vast amount of accumulated experience, points out that the executives must have " a practical knowledge of how to observe, record, analyze and compare essential facts in relation to ...... all...... that enters into or affects the economy of production and the cost of the product," and makes a plea for the interchange of management data.

33 Eight following papers, Nos. 256, 341, 449, 596, 647, 928, 965 and 1012, deal with methods of wage payment, showing the increasing attention given to the workmen during this period. Of these methods the " premium plan " described in paper No. 449 has an extensive use today in machine shops. It probably ranks third, in extent of use, being exceeded by day work and piece work in the order named. Paper No. 647 outlines elementary rate fixing; that is, the minute study of each detail of each operation. From this, motion study and time study have grown. The " bonus system " of paper No. 928 also has an extensive use, probably ranking fourth.

34 Paper No. 1003, " Shop Management," is the first complete presentation of the subject. This paper with the subsequent writings of its author, stands today as the only comprehensive outline of industrial management. Papers Nos. 1001, 1002, 1010, 1011 and 1115, are amplifications of certain features of No. 1003 and are based on the same practice.

35 Paper No. 1221 deals with the training of workmen, and outlines practical, tested methods of bringing about the all-important transference of skill.

LABOR-SAVING MANAGEMENT

36 Since these papers were presented, and during the development of popular interest in the subject, the term " scientific management " has been generally and loosely applied to the new system and methods. This is commonly taken to mean that there is a science rather than an art of management. A truer interpretation is that it means management using scientific methods, these being taken largely from the sciences of physics and psychology.

37 The expression " labor-saving management " better con-

veys the meaning of the movement. It has the further advantage of being easily and surely understood because of its strict analogy with the term "labor-saving machinery." It is no chance that puts these two terms labor-saving machinery and labor-saving management, in conjunction, for the first is the past development and the second the present trend of industry, and they will be closely and inevitably associated in the successful manufacturing of the future. Throughout the following pages of this report the terms "industrial management" and "labor-saving management" are used, the first to denote the subject broadly, the second the newer attitude.

THE REGULATIVE PRINCIPLES OF INDUSTRIAL MANAGEMENT

38 The lack of accurate thinking and clear expression in regard to management are nowhere better shown than in many of the statements of the so-called principles. These can be divided into two classes, personal characteristics of managers and mechanical means of applying. It is evident that neither can show us the way in which the activities of industry are to be regulated.

39 In our investigation preparing for this report, one correspondent writes as follows:

> "The regulative principles of management along scientific lines include four important elements:
>
> *a* Planning of the processes and operations in detail by a special department organized for this purpose.
>
> *b* Functional organization by which each man superintending the workman is responsible for a single line of effort. This is distinctly opposed to the older type of military organization, where every man in the management is given a combination of executive, legislative and judicial functions.
>
> *c* Training the worker so as to require him to do each job in what has been found to be the best method of operation.
>
> *d* Equable payment of the workers based on quantity and quality of output of each individual. This involves scientific analysis of each operation to determine the proper time that should be required for its accomplishment and also high payment for the worker who obtains the object sought."

40 Another correspondent finds the solution of problems of management in the observing and regulating of three classes of industrial phenomena:

> "*a* The economic results of different arrangements and forms of

materials and operations upon them, either to produce equipment or product. This covers the whole field of recorded experience from invention and design of product and tools down through the successive shop processes to ultimate finished product and its tests in service. It is the object of the scientific method to make the best of this experience, in its essential details, readily available for all concerned, and to see that it is actually absorbed and put in practice.

b The economic results of varying executive methods for effectively directing human efforts as a whole in the use of the above experience. This covers the entire field of building up, coördinating and controlling the supervising organization of a plant with its statistical and recording systems.

c The economic results of steps taken to raise the industrial efficiency of the individual worker in every grade of service. This covers the whole problem of labor reward, intensified ability, conserved energy and the general relations of employer and employée."

41 We have pointed out that the underlying principle, that is, cause in the widest sense, the application of which has built up modern industry, is the transference of skill. This basic principle is put into effect on the management side of all industrial activities, through three regulative principles which sum up the ideas in the above quotations, Pars. 39 and 40. These have been concisely stated as[1]: (*a*) the systematic use of experience; (*b*) the economic control of effort; (*c*) The promotion of personal effectiveness.

42 The first includes the use, in all essential detail, of traditional knowledge, personal experience and the results of scientific study on the part of the executive force. It implies the accumulation and use of records and the setting up of standards.

43 The second includes the division and subsequent coördination of both executive and productive labor; the planning of single lines of effort, the setting of definite tasks and the comparison of results; and the effective training of the workers. It implies the previous acquisition of skill by the executives.

44 The third includes a definite allotment of responsibility and the adequate, stimulative encouragement and reward of both executive and productive labor; the development of contented workers, and the promotion of their physical and mental health. It implies the most thorough comprehension of the human being.

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[1] American Machinist, vol. 36, p. 857, The Principles of Management, by Church and Alford.

THE PRACTICE OF MANAGEMENT

45 As labor-saving management springs from a change in mental attitude, the beginning of its practice should be with the persons having the final responsibility, the proprietors of closely-owned businesses, the directors of larger establishments, or the officials having charge of Government works. Before any changes are made, such men should clearly understand the viewpoint from which all of the managerial work is to be done, the principles that are to be applied, the general method of their application and the results expected.

46 A similar mental attitude must be fostered among all the members of the executive force and a period of training for them begun. This may include a redistribution of function and responsibility, and will include a detailed study of production by scientific methods. This is the period of division of thought, training of the management staff and setting-up standards of performance. This must be carefully performed before there can be effective transference of skill to the workers in the production departments.

47 The usual conception of modern management is that it affects the workmen most of all, tending to stimulate them to turn out increased production to their possible hurt. This is wrong. If the principles outlined are followed, the executive, or non-producing labor is the most affected. Its individuals are compelled to study, plan and direct. They must acquire knowledge and skill in order to transfer it. It is a system of management that forces the executives to manage.

48 This being so, the introduction of modern management in a plant must be made slowly. The causes of most so-called failures are principally two: a failure of the executives to acquire the vital mental attitude and too great haste in application. The latter seems to be the dominant one. Your committee feels compelled to emphasize the danger of attempting to hurry any change in methods of management. Each step of the work should be made permanent before the next is begun.

49 We have examined records of production which clearly show a lessening of individual output among workers who had been trained for some time and had achieved good results as soon as untrained workers were put with them, thus lessening their share of personal supervision. Later the original standard

of production was again reached, but the results seemed to be directly proportional to the amount of skilful supervision, during a lengthy period of training.

50 After those who are to operate the new methods have acquired the necessary knowledge and established sufficient standards, the work of putting these into effect can be begun. This means the fixing of the best attainable working conditions and giving each worker definite tasks with an adequate reward to each one who attains to the standard set. This part of installing the methods must be accomplished with tact and patience, remembering that leadership and example are powerful aids in bringing about enthusiastic coöperation.

51 The training of the workers is essential in this part of the application. This must be far more than mere demonstration, the mere showing that a thing can be done. It must be patient teaching and help until the required degree of dexterity or skill is acquired, that is, up to the habit stage. It is evident that such work cannot be hurried.

52 Such, broadly, are the three steps in the practice of management. It is now necessary to investigate the internal elements of permanence in such methods. If the proper mental attitude is once taken, we believe it will never be given up. This is substantiated by a few cases when early attempts to improve management were failures and the methods abandoned. Later, however, other attempts were made with substantial success. The mental attitude outlived the failure. Thus in a given industrial organization this feature would not be lost except by a loss of the executive staff.

53 The permanence of records of performance and standards needs only to be mentioned to be appreciated. Once set up in an industry, disaster is invited if they are disregarded.

54 To these is added a third in the nature of a spur from the working forces to the managing force. An adequate reward is one of the essentials. Whatever disturbs the mechanism of production interferes with the earning of the rewards. The workers at once object, pointing out the trouble and insisting that it be rectified. The management is spurred to keep all conditions up to the fixed standard. Examples of this action have been brought to the attention of your committee.

55 The practice as outlined, while built upon fixed standards

and procedure, is by no means rigid and inflexible as has been alleged. The design and construction of labor-saving machinery is carried on with a multiplicity of different details. Labor-saving management should likewise use a variety of details suited to the requirements of different industries and plants. There can be nothing fixed in such human endeavor except the underlying principle. As a simple matter of fact we have found different methods, details and nomenclature in use in different plants. Many efforts have undergone marked change and development since first installed. Further, this idea of rigidity is repudiated by some of the foremost management experts.

56 In Par. 39 is emphasized the need of a scientific study of everything connected with production. The methods used are adapted from the research laboratory. But the purpose of their use is changed. The scientific investigator uses his laboratory to discover facts. Their discovery and declaration is his end and aim. The management investigator uses laboratory methods to discover facts for immediate use. The end and aim is utility. This is the test of industry. It is therefore unwise and in fact detrimental to carry investigations to an extreme. Enough facts must be observed to shape intelligent action. Persons having time study and motion study in charge should possess that rare, intuitive, human quality that causes its possessor to know when enough observations have been collected to form a sound working conclusion.

57 The position of the expert in the practice of management is more clearly seen as experience increases. The element of mystery has already departed. This is to be welcomed for it means the downfall of mere "systematizers." One of the unfortunate features of this great movement has been the rise of alleged experts who have been ready to promise extravagant results if they were allowed to systematize an industrial plant. The test which their work cannot meet is the one of permanence.

58 An industrial manager who has had signal success in directing large enterprises sums up the more undesirable characteristics of systematizing practice as:

> "*a* The publication and quotation of statistics regarding gains made through the use of particular systems, without a frank statement of the degree of inefficiency of the plants before reorganization.

b The failure to view the plant from the investor's standpoint rather than as a laboratory offering opportunities for interesting and expensive experience.

c The failure to admit that every application of past solutions to unstudied new and different conditions *is* an experiment.

d The waste of time and money on problems that will yield to scientific treatment, but which do not recur often enough to justify such a solution.

e The undervaluing of effective leadership in management and consequent lack of permanency in results.

f The overvalue of emasculated 'system' leading to a curious non-responsibility on the part of any person for the total result.

g The frequent assumption that the treatment of the problems of similar plants should be identical.

h The failure to properly appraise in a growing concern the value of its internal asset of 'good will.'

i The imperfect analysis and appreciation of the human factor in industry, with a consequent failure to reckon patiently with 'habit' and 'inertia' and a tendency to hasty 'substitution,' bringing about the breaking up of valuable organization."

59 The real expert concentrates on the facts of a given problem, and from a wide experience in analysis, coördination and practical responsibility works out a solution by scientific methods, suited to the material and human factors involved. The tendency is for him to do less of the detail work of installation, but to train and direct the persons who are permanently to manage. This is a true process of transference of skill.

STATISTICAL DATA

60 Your committee hoped to present statistics on the extent to which labor-saving management is in use. This could not be realized. Many industrial managers whom we have addressed have not honored us with their confidence in this direction. In fact, it seems as if a secretive stage is now with us. There are two reasons for withholding such information. The first is identical with the one that has developed "trade secrets" and secretiveness in regard to machines, tools and processes, the desire to keep things of value away from competitors. The second is a belief that in the minds of some persons a reflection is cast upon the ability of the executives of an industrial establishment if outside experts are employed. Frequently a system of management is referred to as the development of some one in the organization, although it was installed by a management expert, employed for the purpose.

61 Some idea of the variety of the industries in which labor-saving management is in use can be gained from Appendix No. 3, which lists a total of 52.

BROAD RESULTS OF LABOR-SAVING MANAGEMENT.

62 In cases where the use of labor-saving management can be considered a success, the broad results have been: a reduced cost of product; greater promptness in delivery with the ability to set and meet dates of shipment; a greater output per worker per day with increased wages; and an improvement in the contentment of the workers. This last item is shown by the fewness of strikes under the new management, and in the refusal of those working under the changed conditions to join in a strike of their fellows in the same plant who were not working under the new methods. This last-mentioned situation has arisen a number of times. In one case an attempt was made to strike a room where about one-half of the operators were under the new conditions. These refused to go out; the rest went.

63 These results indicate certain advantages to both employer and employé. But it is charged that the movement has not yet entirely justified itself from the economic viewpoint, for it has not reduced the cost of product to the consumer. The implication is that its possibilities will not be realized until employers, employés and the public are alike benefited. With this view we are in most hearty accord. Labor-saving machinery has brought the comforts that we all enjoy today. Labor-saving management promises to extend those comforts. Where properly administered it is conserving labor and is thus contributing to the good of society at large, and although the benefit to the consumer may not yet be generally felt, it has already developed to a certain extent and will continue to develop as the natural result of increased production.

J. M. Dodge, *Chairman*
L. P. Alford, *Secretary*
D. M. Bates
H. A. Evans
Wilfred Lewis
W. L. Lyall
W. B. Tardy
H. R. Towne

Members Sub-Committee on Administration

APPENDICES

APPENDIX NO. 1

64 The following tables are taken from a book published in 1832, Economy of Machinery and Manufacture, by Charles Babbage. Table 1 gives the cost and operations in detail in manufacturing 12,000 No. 6 pins in France in 1760. The observations were made by M. Perronet.

65 Table 2 gives similar data for manufacturing pins in England in the time of the author. The size is "eleven" of which there are 5546 to the lb.

TABLE 1 OPERATIONS IN PIN MANUFACTURE IN FRANCE IN 1760

	Name of the Process	Time of Making 12,000 Pins, Hr.	Cost of Making 12,000 Pins, Pence	Workman usually Earns per Day, Pence	Expense of Tools and Materials, Pence
1	Wire	....	....	....	24.75
2	Straightening and Cutting	1.2	0.5	4.5	....
3	Coarse Pointing	1.2	0.625	10.0	....
	Turning Wheel *	1.2	0.875	7.0	....
	Fine Pointing	0.8	0.5	9.375	....
	Turning Wheel	1.2	0.5	4.75	....
	Cutting off Pointed Ends	0.6	0.375	7.5	....
4	Turning Spiral	0.5	0.125	3.0	....
	Cutting off Heads	0.8	0.375	5.625	....
	Fuel to Anneal Heads	....	....	....	0.125
5	Heading	12.0	0.333	4.25	....
6	Tartar for Cleaning	....	....	....	0.5
	Tartar for Whitening	....	....	....	0.5
7	Papering	4.8	0.5	2.0	....
	Paper	....	....	....	1.0
	Wear of Tools	....	....	....	2.0
		24.3	4.708		

* The expense of turning the wheel appears to have arisen from the person so occupied being unemployed during half his time, whilst the pointer went to another manufactory.

TABLE 2 OPERATIONS IN PIN MANUFACTURE IN ENGLAND ABOUT 1830

	Name of the Process	Workmen *	Time of Making 1 Lb. of Pins, Hr.	Cost of Making 1 Lb. of Pins, Pence	Workman Earns per Day		Price of Making Each Part of a Single Pin in Millionths of a Penny
					s.	d.	
1	Drawing Wire	Man	0.3636	1.2500	3	3	225
2	Straightening the Wire	Girl	0.3000	0.1420	0	6	26
		Woman	0.3000	0.2840	1	0	51
3	Pointing	Man	0.3000	1.7750	5	3	319
4	Twisting and Cutting the Heads	Boy	0.0400	0.0147	0	4½	3
		Man	0.0400	0.2103	5	4½	38
5	Heading	Woman	4.0000	5.0000	1	3	901
6	Tinning, or Whitening	Man	0.1071	0.6666	6	0	121
		Woman	0.1071	0.3333	3	0	60
7	Papering	Woman	2.1314	3.1973	1	6	576
			7.6892	12.8732			2320

* Number of Persons Employed: Men, 4; Women, 4; Children, 2. Total, 10.

APPENDIX NO. 2

66 Following is a complete list of papers published in the Transactions of The American Society of Mechanical Engineers dealing with industrial management:

No.			
207	The Engineer as an Economist	Henry R. Towne	1886
256	A Problem in Profit Sharing	Wm. Kent	1887
341	Gain Sharing	Henry R. Towne	1889
449	The Premium Plan of Paying for Labor	F. A. Halsey	1891
596	The Relation of the Drawing Office to the Shop in Manufacturing	A. W. Robinson	1894
647	A Piece-Rate System	Fred. W. Taylor	1895
928	A Bonus System for Rewarding Labor	H. L. Gantt	1902
965	Gift Propositions for Paying Workmen	Frank Richards	1903
1001	The Machine-Shop Problem	Charles Day	1903
1002	A Graphical Daily Balance in Manufacture	H. L. Gantt	1903
1003	Shop Management	Fred. W. Taylor	1903
1010	Slide Rules for the Machine Shop as a Part of the Taylor System of Management	Carl G. Barth	1904
1011	Modifying Systems of Management	H. L. Gantt	1904
1012	Is Anything the Matter with Piece Work?	Frank Richards	1904
1115	A History of the Introduction of a System of Shop Management	James M. Dodge	1906
1221	Training Workmen in Habits of Industry and Coöperation	H. L. Gantt	1908

APPENDIX NO. 3

67 Following is a list of the industries in which some form of labor-saving management has been installed:

- Book binding
- Building construction
- Carriage and wagon building
- Construction and repair of vessels (navy yards)
- Fire-arms and ordnance
  - Rifles
  - Gun carriages
- Machinery building
  - Automobiles
  - Agricultural implements
  - Coal-handling machinery
  - Electrical machinery
  - Founding, iron and brass
  - General machine work
  - Gas engines
  - Locomotives
  - Machine tools
  - Molding machines
  - Pumps
  - Pneumatic toools
  - Sewing machines
  - Typewriters
  - Wood-working machinery
- Metal and coal mining
- Metal working
  - Bolts and nuts
  - Chains
  - Hardware
  - Tanks
  - Tin cans
  - Valves and pipe fittings
- Miscellaneous manufacturing
  - Beer
  - Beet sugar
  - Boxes (wood and paper)
  - Buttons
  - Clothing
  - Cordage
  - Food products
  - Furniture
  - Flour
  - Glass
  - Lumber products
  - Pianos
  - Paper and paper pulp
  - Rubber goods
  - Soaps
  - Shoes
  - Slate products
- Printing and lithographing
- Railroad maintenance of motive power
- Steel manufacture
- Textile manufacture
  - Bleaching and dyeing
  - Cottons
  - Velvets
  - Woolens

THE PRESENT STATE OF THE ART OF INDUSTRIAL MANAGEMENT

MINORITY REPORT OF SUB-COMMITTEE ON ADMINISTRATION

I am unable to sign the majority report in its entirety, much as I admire the thoroughness with which it has been prepared and its great interest. In its general tenor it distinctly implies the desirability of what is termed labor-saving management, involving the planning department, functional organization and the bonus system. Perhaps this statement is not strictly justified, but I cannot avoid the impression after reading the report most carefully. That the methods of management have undergone a great change in recent years I certainly agree. I would explain it by stating that in many respects the art of management is developing into a science. In common with most lines of work, the method of investigating facts has changed. Phenomena are analyzed, information is obtained accurately, the mental attitude of the manager is scientific rather than empirical. Things that used to be known generally by gradual experience are now known specifically by detailed observation. In the course of this development many new systems and ideas have been invented, time studies, motion study, payment systems, functional management, etc. Some are new, others partly new, others simply practices of many managers put into definite form. The science of management will classify these for us, perhaps even explain their advantages and limitations so that we may in time know which is preferable and when.

2 But the introduction of the use of these methods has been attended by claims as to the results that might be obtained by their use, and these claims have led to such absurd statements as that made before the Interstate Commerce Commission, which is referred to in the report. I feel most strongly that each of the suggestions that have been made to improve methods of management may have merit, but I do not feel that any one of them is a panacea for all our inefficiency. For instance, in certain classes of work time studies are valuable, in others they may be a waste of time. In certain

classes of work I consider piece-work or bonus systems desirable, in others I consider them inferior to day work. Functional management may be an improvement in certain industries, in others I do not consider it suitable. In some cases the workman can be trained by the skilled executive, in others he can train him with ease.

3 In general I feel that labor-saving management is not any particular system, but will always remain the art of selecting and applying the most appropriate methods furnished by the science of management, the science that records what these methods are and the results obtained from them. As you will see, I agree with your correspondent in his last paragraph, quoted under Par. 28; but his opinion is not really represented in the trend of the report.

H. H. Vaughan
Member, Sub-Committee on Administration

AIR IN SURFACE CONDENSATION

By Geo. A. Orrok

ABSTRACT OF PAPER

This paper is an addition to the author's paper on the Transmission of Heat in Surface Condensation and gives the results of his investigations on the amount of air present in feedwater and the presence and effect of air in surface condensers, and describes an apparatus for measuring the air leakage.

AIR IN SURFACE CONDENSATION

By Geo. A Orrok, New York

Member of the Society

In the paper on The Transmission of Heat in Surface Condensation[1], the subject of the effect of air on vacuum and heat transference was considered and reference was made to the few authorities who had investigated this subject. The author was unable to find that any one had quantitatively measured the air in a condenser, except J. A. Smith who had introduced known quantities of air and noted the effect on vacuum and heat transference.

2 In view of the lack of knowledge of this subject the author determined to investigate:

a The amounts of air contained in feedwater and those entering the condenser through the prime mover.

b The amounts of air removed from the condenser by means of the air pump.

c The effect of known quantities of air on the action of a surface condenser of commercial size.

3 The problem of air in the feedwater has been investigated by Smith whose paper appeared in London Engineering, October 7, 1904. He gives a table showing that the gases present in various samples of fresh and salt water varied from approximately 7/10 of one per cent to as much as 3½ per cent by volume. He also investigated the emission of gas from water when heated to about the boiling point and gave figures and curves showing his results.

4 The author's investigations were made at the Waterside No. 2 station of the New York Edison Company and the water samples were taken both from the suction and discharge sides

[1] Trans. Am.Soc.M.E., vol. 32, p. 1139.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

of the feed pump. The average temperature of the feedwater during the tests, after having passed the heater, was about 190 deg. fahr. It was thought possible that there might be air bubbles mechanically entrained in the water, besides the amount of air in solution. To determine the amount of this entrained air samples of the water were allowed to flow through the apparatus which is shown in Fig. 1. The water entered through the tube *F* and passed to the top of the measuring cylinder where the air separated and was collected at the top of the cylinder; the water flowed out of the cylinder through the tube *B* into the glass demijohn where it was measured. These tests were started with the apparatus entirely filled with water. The air collected was measured at atmospheric pressure by means of the pressure-regulating cylinder *D*. The temperature of the air was determined by the thermometer *E*. The amount of air dissolved in the feedwater was determined chemically by means of the following reactions:

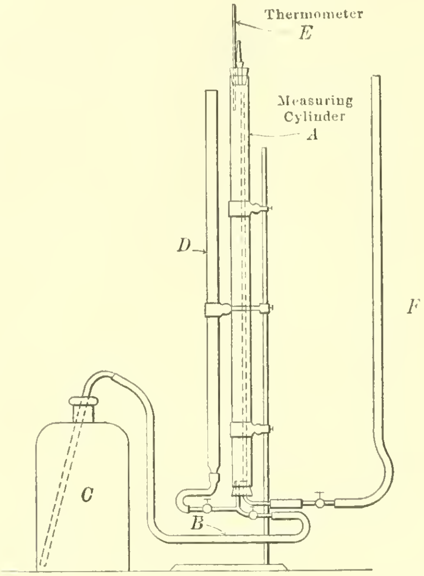

Fig. 1 Apparatus used to determine Amount of Air Mechanically entrained in Water

$$2Mn(OH)_2+O_2=2H_2MnO_3$$
$$2H_2MnO_3+8Hcl=2MnCl_2+6H_2O+2Cl_2$$
$$4KI+2Cl_2=4KCl+2I_2$$

5 The free iodine liberated was titrated with a solution of sodium thiosulphate of known strength and its amount determined. From this amount the quantity of oxygen in solution in the sample of water taken was calculated. The amount of air in solution was then figured from the known ratios of oxygen and nitrogen in solution in water at various temperatures. The determination in this way may be considered quite accurate.

6 This determination of air in water was also made of a number of samples of Croton water (the make-up feed) and from hot well water. The following results, in per cent, were obtained of air by volume at atmospheric pressure:

	187 deg. fahr.
Mechanically entrained in feedwater, average of ten tests	0.0151
In solution in feedwater, average of ten tests	0.916
Total air in feedwater from open heaters	0.931
	52 deg. fahr.
In solution in Croton water, average of three samples	4.325
	80 deg. fahr.
In solution in hot well water, average of four tests	0.269

7 These results show that Croton water gives off most of its dissolved air in passing through the open heaters, the amount of air occluded in the feedwater (0.931 per cent) being comparatively small. At 81 deg, fahr. and 2 in. of mercury absolute pressure when saturated with water vapor, this amount of air would occupy only 1/17 of the volume of the condensed steam.

8 The results obtained accord well with reports from other sources, although the author could find but little quantitative information on the subject. The fact was established, however, that very little air came into the condenser from the feedwater.

9 Having obtained the portions of air which enter the prime mover along with the steam, the next object of investigation was to determine the amount of air abstracted from the condenser by the air pump. Here again there were no precedents but it was determined to catch the air over water in a gasometer. For this purpose the discharge pipe of the air pump was piped to a

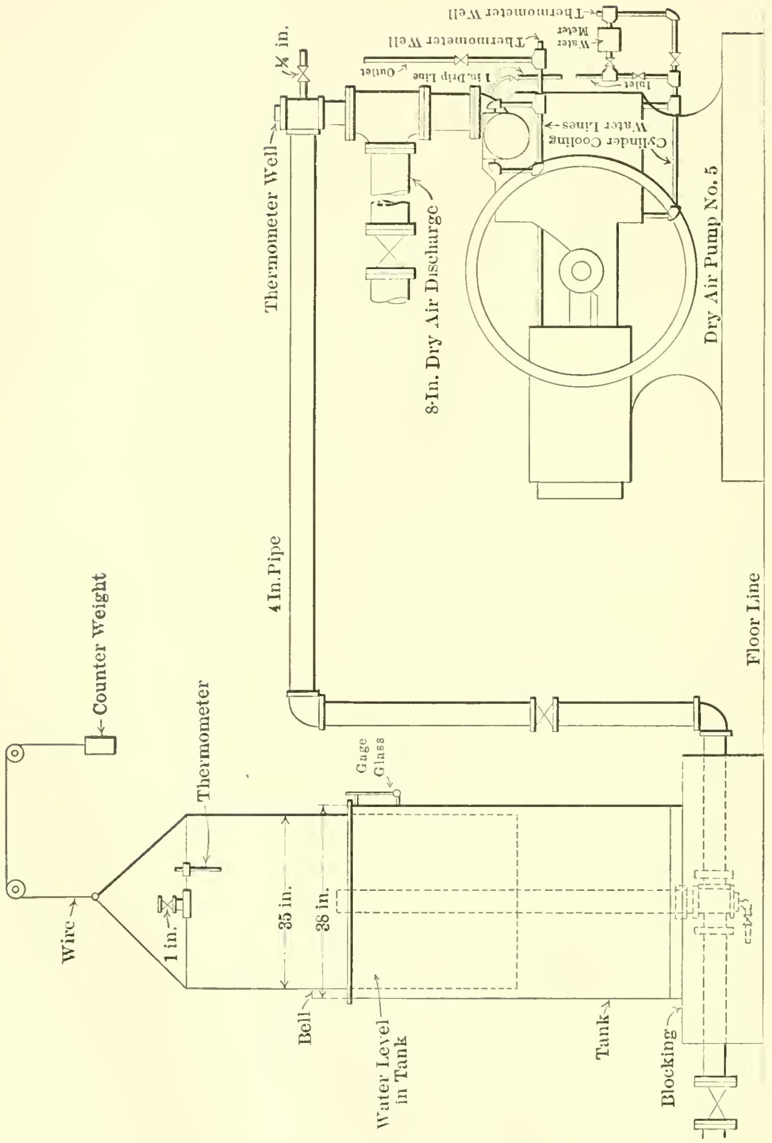

FIG. 2 LAYOUT FOR MEASURING AIR DISCHARGED BY DRY VACUUM PUMP

gasometer bell located in the basement in the vicinity of the air pump. A sketch plan of this outfit is shown in Fig. 2.

10 After the air pump had been running for a sufficient time to make sure that the conditions were constant, the valve was opened and the discharge from the air pump was allowed to enter the bell allowing it to rise. The time required to fill the bell was noted with a stop watch and from these data the amount of air delivered was calculated. The air in the bell was brought to atmospheric pressure and temperature before the final readings were taken.

11 Other tests were made with the pump shut off from the condenser to determine the air leakage in the pump itself and also with the suction piping blanked off at the condenser to determine the pipe leakage.

12 In a number of cases these readings showed excessive leakage in the condenser shell which was remedied before the final readings were taken. A second apparatus has been provided for Waterside No. 1 and both stations at the present time are so arranged that in a few hours the air leakage of all the air pumps as well as the condensers may be tested. This is a great help in operating and serves a useful purpose in enabling attention to be given where there was no indication formerly of leakage.

13 A great many air tests have been made on all of the units in the two stations. Usually readings were taken during the time necessary to fill the bell (from 1 to 20 minutes) of the air temperature in the bell, the air suction temperature, air discharge temperature, vacuum at the top of the condenser, vacuum in the air suction line to the pump, hot well temperature, circulating water inlet and outlet temperature, amount of cylinder cooling water with its inlet and outlet temperatures, revolutions per minute of the pump, power delivered by the prime mover and sometimes indicator cards on both air and steam cylinders. Table 1 shows a collection of some of the more interesting of these tests.

14 In the experimental apparatus used to obtain the data for the paper on The Transmission of Heat, it was impossible to detect any difference of temperature due to the segregation of air in the various parts of the single tube condenser. On the large condensers on which these air tests were made such

TABLE 1 RESULTS OF TESTS ON DRY AIR PUMPS FOR SURFACE CONDENSERS

																		Water Used to Cool Air Cylinder		
Number of Tests Made	Date of Tests	Load on Turbine, Kw.	Vacuum at Top of Condenser Corrected to 30 In. Barometer	Vacuum of Pump Suction Corrected to 30 In. Barometer	Barometer	Hot Well	Circulating Water Inlet	Circulating Water Outlet	Air Suction	Air Discharge	Air in Bell	R.P.M. of Pump	Air discharged per Minute measured in Bell	Air discharged per Minute at Temperature of Air Discharge	I.H.P. Steam Cylinder	I.H.P. Air Cylinder	Mechanical Efficiency of Pump, Per Cent	Temperature Inlet, Deg. Fahr.	Temperature Outlet, Deg. Fahr.	Cubic Feet used per Minute
			Inches of Mercury			Temperatures, Deg. Fahr.							Cubic Feet							
3	2/14/11	6,500	28.49	28.61	30.09	54.7	35.7	48.8	63.0	144.7	94.3	70.0	12.94	17.15	26.92	14.80	57.1	....	46.3	....
5	2/14/11	8,200	28.43	28.61	30.07	55.0	33.0	49.8	68.0	141.7	93.3	61.4	8.59	11.08	22.85	10.36	45.7	....	52.6	....
5	2/14/11	10,400	28.16	28.40	30.04	63.0	33.0	54.2	81.2	139.4	93.1	60.4	5.82	7.40	21.52	10.24	47.6	....	53.7	....
5	2/18/11	6,266	28.50	28.60	29.92	53.0	33.6	47.9	62.8	138.2	96.5	67.2	12.65	15.65	22.51	11.94	53.0	38.56	45.0	....
5	2/18/11	8,133	28.46	28.60	29.92	57.8	33.4	51.1	73.4	136.2	96.0	66.4	6.86	8.50	21.80	9.35	42.9	38.00	45.6	2.21
5	2/18/11	10.132	28.11	28.34	29.92	68.0	33.0	54.2	84.9	137.0	95.5	65.4	5.75	7.12	20.08	10.85	58.9	38.00	46.0	2.19
3	2/18/11	10,000	28.08	28.32	29.90	68.3	33.3	54.7	85.2	144.8	97.3	74.0	5.93	7.83	23.08	12.30	53.3	37.20	43.0	3.91
2	2/18/11	10,000	27.95	28.27	29.90	69.0	34.0	56.0	86.0	144.5	96.3	72.5	5.89	7.78	23.89	11.75	49.2	38.30	49.8	....
2	2/18/11	10,000	27.90	28.23	29.90	69.5	34.5	56.0	87.7	147.3	96.0	72.0	5.27	7.07	23.32	10.58	45.3	40.00	67.8	0.49
*3	2/28/11	No load	....	29.30	30.20	....	....	....	....	99.3	98.6	75.6	1.31	....	25.32	13.99	55.3	37.70	43.0	3.14
4	3/14/11	6,075	....	29.00	30.39	....	....	....	55.0	114.0	83.5	77.0	8.28	....	....	....	....	....	....	....
6	3/14/11	9,252	....	28.99	30.35	....	....	....	57.0	111.2	88.2	75.5	6.94	....	....	....	....	....	....	....
4	3/14/11	11,613	....	28.93	30.31	....	....	....	69.8	118.9	85.5	75.5	6.33	....	....	....	....	....	....	....
*2	3/15/11	No load	....	29.10	29.98	....	....	....	....	85.7	96.5	76.5	0.63	....	....	....	....	....	....	....
	1/26/12	9,100	27.67	28.74	29.89	....	33	60	91.8	105.0	87.2	....	29.25	31.25	....	....	....	....	....	....
	1/26/12	5,970	28.15	28.65	30.03	....	33	51	91.0	100.5	88.0	....	29.60	30.95	....	....	....	....	....	....
6	4/24/11	3,022	....	28.05	30.25	....	....	....	59.1	184	95	81	39.5	....	....	....	....	....	....	....
4	4/24/11	5.204	....	28.00	30.18	....	....	....	60.6	198	98	80	38.5	....	....	....	....	....	....	....
6	4/20/11	8,258	....	28.10	29.78	....	....	....	59.0	164	93	85	36.8	....	....	....	....	....	....	....

*Valve in suction line closed.

a difference of pressure and temperature does exist, the differences of pressure being quite strongly marked. The curves for a number of these condensers, showing the increase in drop with the increase in amount of steam consumption, are illustrated in Fig. 3.

15 A number of experiments were made to find if this drop was continuous through the tube surface or whether there were restrictions concentrating the drop in one or more places, and a number of experiments were made to reduce the amount of drop by taking out tubes thus opening channels into the body of the tube surface. The results here were apparently negative.

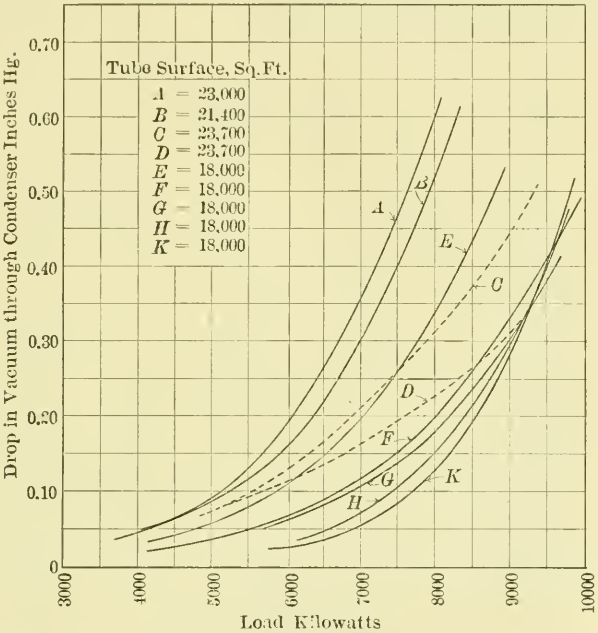

FIG. 3 CURVES SHOWING RELATION BETWEEN LOAD AND DROP THROUGH CONDENSER

16 To ascertain the effect of known quantities of air on the vacuum suitable orifices of various sizes were installed at the top of the condenser so that the introduced air might mix with the incoming steam. These orifices were the ordinary circular orifices through thin plates and the amounts of air entering the condenser were checked up by the air delivered into the

air bell from the air pump. These results are shown in Fig. 4.

17 The tests in each case were run at constant load which was considerably below the capacity of the condenser.

18 Four sets of tests were run to determine the effect of load on air leakage, two on machines which were in bad condition and two on comparatively tight machines. The results are shown in Fig. 5. It should be noted that in every case the air leakage is less at the higher loads.

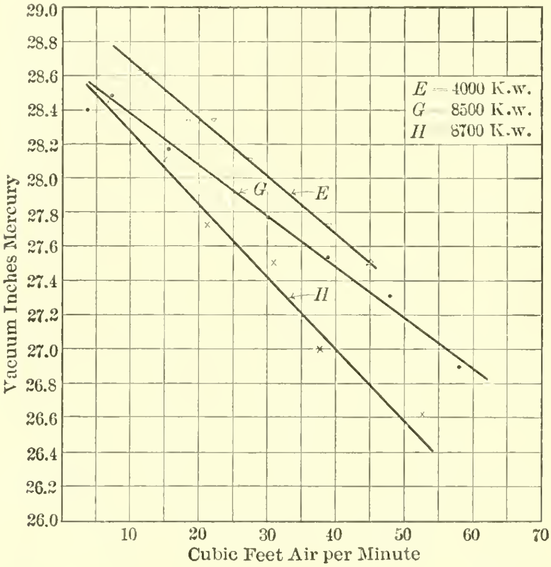

FIG. 4 CURVES SHOWING EFFECT OF AIR LEAKAGE AT CONSTANT LOAD

19 In the paper on The Transmission of Heat it was inferred from Smith's curves that the heat transfer varied as the fifth power of the ratio $\frac{p_s}{p_t}$. Since that time many tests have been run from which sufficient data to calculate this relation were obtained, and the results are shown in Fig. 6. None of the tests approach the capacity of the tube surface but the results are consistent.

CONCLUSIONS

20 Where open heaters are used very little air is carried to the prime mover with the steam. The volume of this air at at-

mospheric pressure and temperature will probably not exceed 1 per cent of the volume of the feedwater. As the whole system is under pressure from the pump nearly to the final stage of the turbine no air leakage can take place up to that point.

21 The air discharged by the dry air pump at atmospheric pressure and temperature from units between 5000 kw. and 20,000 kw. in size varies from 1 cu. ft. per min. when the units are in the best condition, to 15 or 20 cu. ft. where ordinary leakage is present, and to 30 or perhaps 40 or 50 cu. ft. where the units are in very bad condition.

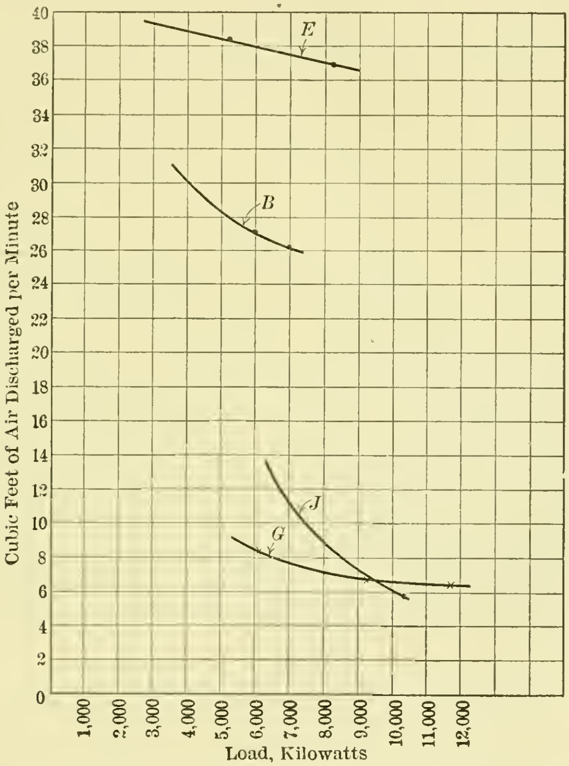

FIG. 5 CURVES SHOWING EFFECT OF LOAD UPON AIR LEAKAGE

22 Most of this leakage comes into the condenser and exhaust passages through minute leaks in the cast-iron shells, gaskets, and expansion joints. It is exceedingly difficult to detect these leaks with a candle flame but most of them may be detected by

filling the condenser with warm water under a slight head. That portion of the leakage occuring in the dry air pump and piping may be detected by the shut-off test on the pump. The volume of this leakage is larger than generally supposed and is much larger when the pump is warm.

23 The drop in vacuum through condensers of standard design is larger than it should be and a design which freely admits the steam to the tube surface should give better results. The opening of channels into the surface will not always improve conditions. The results indicate that wide shallow condensers should give better results although in at least two instances deep narrow condensers gave good results.

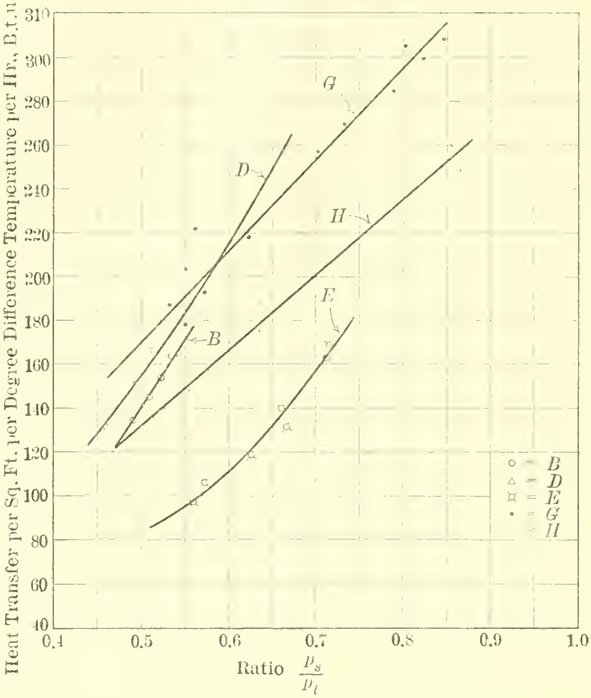

Fig. 6 Curves showing Relation between Ratio $\frac{p_s}{p_t}$ and Heat Transfer

24 The falling off of vacuum with increased air leakage is considerable and the work necessary to maintain tight condensers more than pays for itself.

25 Air leakage decreases with an increase of load. This fact in some measure corrects the decrease of vacuum at the increased loads.

26 The exponent of the ratio $\frac{p_s}{p_t}$ on page 1161 of Volume 32 of the Transactions of the Society may be changed from 5 to 2 pending further experiments to establish a better value.

27 During these experiments in which four different kinds of dry air pumps were investigated it became evident that the volumetric efficiency of a pump working between ¼ lb. and 15 lb. absolute is very poor indeed. The difficulties of keeping valves, pistons, glands and packing tight against such a tenuous fluid as air at vacuum pressures are of much greater moment than is usually believed. It is hoped that some of the types of kinetic air pumps now being placed on the market will reduce these difficulties to a practical amount.

28 Considerable difficulty was experienced in securing vacuum temperatures during these experiments and most of the work has been repeated a number of times where such temperatures are essential. It has been impossible up to the present time to secure readings which were entirely consistent and the work along this line will be carried on until the cause of the discrepancy is ascertained and correct figures obtained.

29 The author wishes to acknowledge the efficient aid rendered him in conducting these experiments by his assistants, Messrs. E. B. Rickets, J. H. Lawrence and J. S. Kerins, and by P. E. Reynolds, engineer of tests.

MEASUREMENT OF NATURAL GAS

By Thos. R. Weymouth

ABSTRACT OF PAPER

In the measurement of natural gas there are two distinct requirements to be met. The first relates to the determination of the open flow capacity of gas wells, for which two principal methods are used, namely, the "minute pressure" measurement and the pitot tube. This paper describes these methods, giving the proper formulae to be used with them.

The second requirement is that of the measurement of flow in closed pipes. While large proportional meters are used extensively for this purpose, the high pressures often encountered have a tendency to introduce serious errors, making it necessary to resort to more accurate devices. Of these, the best known are described, namely, the Thomas electric meter, the venturi meter, the pitot tube, and the orifice. The general formulae are given for the last three named, together with their derivations.

MEASUREMENT OF NATURAL GAS

By Thos. R. Weymouth, Oil City, Pa.

Member of the Society

One of the most important problems encountered in natural gas engineering is the accurate measurement of gas in large volumes. For the purpose of discussion, the subject can be divided into two general classifications, namely, open flow and closed flow measurements. The first relates principally to the determination of the volume of gas a well is capable of producing, while the second refers to all closed pipe flows.

2 Under the first head, there are two general methods in use for measuring the open flow of a well, that is, the quantity of gas the well is capable of producing against an absolute back pressure of one atmosphere. The first of these is purely approximate, and is called the "minute pressure" measurement, wherein the well is permitted to blow into the air until its "head" is blown off, and is then suddenly closed in by a quick-acting gate, the accumulated pressure being noted at the end of one minute from the time the gate is closed. Then, Q, the open flow capacity of the well in cubic feet per day, on an atmospheric pressure base of 14.4 lb. per sq. in. absolute, becomes

$$Q = 1440\ V\left\{\frac{p_m + P_o}{P_o} - \frac{P_o}{P_o}\right\} = 1440\ V\frac{p_m}{P_o} = 100\ Vp_m \qquad [1]$$

in which

V = volume of well tubing in cu. ft.

p_m = minute pressure in lb. per sq. in. gage

P_o = atmospheric pressure in lb. per sq. in. absolute

3 This equation assumes that while the gas is flowing freely into the air, the well is filled with it at a uniform pressure of one atmosphere, and at the end of one minute it is filled with gas at a

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uniform pressure equal to the minute pressure. Neither of these assumptions is correct, since in both cases there is a gradually increasing pressure from the top of the well towards the bottom, similar to that in a pipe line.[1] The resulting errors tend to counteract each other, however, and if care is taken in making the measurements a comparative idea will be obtained of the ability of the well to produce gas against a back pressure equal to a mean between the atmosphere and the minute pressure. Since the open flow will be greater than the flow against this back pressure, the assumption that the minute pressure gives an open flow measurement errs on the side of conservatism.

4 The second method of open flow measurement is that by means of the pitot tube, and is the one generally used for accurate determinations of well capacities. This method has been described in detail by Prof. S. W. Robinson,[2] who was probably the first to use it in connection with gas. The method was further discussed by him in the Geological Reports of Ohio,[3] wherein tables are published by means of which the flow may be computed from the pitot tube observations. These tables have been in general use by natural gas men for many years, and are based on the formula

$$Q=1{,}462{,}250\ d^2\left\{\left(\frac{P_1}{P_0}\right)^{0.29}-1\right\}^{\frac{1}{2}}\ldots\ldots\ldots\ldots\ldots[2]$$

5 Equation [2] was derived from the formula for adiabatic flow

$$v^2=\frac{2gn \times 144\ P_0}{(n-1)\ \delta}\left\{\left(\frac{P_1}{P_0}\right)^{\frac{n-1}{n}}-1\right\}\ldots\ldots\ldots\ldots\ldots[3]$$

In equations [2] and [3]

v = velocity of flowing gas, ft. per second

P_0 = absolute pressure of atmosphere, lb. per sq. in.

n = ratio of specific heats

δ = weight per cu. ft. of gas at pressure P_0

G = specific gravity of gas, air = 1.0

P_1 = absolute pressure shown by pitot tube, lb. per sq. in.

d = internal diameter of well mouth, in.

Q = open flow capacity of well, cu. ft. per 24 hr.

6 In computing his tables, Professor Robinson used the following values for these quantities:

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[1] Problems in Natural Gas Engineering, The Journal, Am. Soc. M. E., May 1912, p. 723, Fig. 5.

[2] Van Nostrand's Engineering Magazine, August 1886.

[3] Vol. 6, 1888.

$$n = 1.408$$
$$2g = 64.3$$
$$P_o = 14.6$$
$$G = 0.6$$
$$\delta = 0.0807 \frac{T_o}{T} G$$
$$T = T_o = T_s = 492 \text{ deg.}$$

where

T_o = absolute temperature of melting ice
T = absolute temperature of flowing gas
T_s = absolute temperature of storage

all being in fahrenheit degrees.

7 The ratio of specific heats for natural gas is more nearly equal to 1.266 than to the value used above. Inserting this value and using values for the other quantities as follows:

T = 500 deg.
T_s = 520 deg.
P_o = 14.4
Storage Pressure, P_s = 14.65

the formula becomes

$$Q = 1{,}758{,}560\, d^2 \left\{ \left(\frac{P_1}{14.4} \right)^{0.21} - 1 \right\}^{\frac{1}{2}} \ldots\ldots\ldots\ldots\ldots [4[$$

8 At 60 lb. gage pressure per sq. in. (the highest given by Professor Robinson), formula [4] makes the flow about 1 per cent lower than that derived from formula [2], whereas at 1 lb. it gives a result about 4 per cent higher.

9 The open flow measurement is not required with the same degree of accuracy as that of closed flow, where gas is being bought and sold, and at best is only intended to give a fair idea of the production. Even if an accurate measurement were made it would give no precise notion of the real productive value of the well when it must feed into the line against a high back pressure. Consequently, if it is desired to know the capacity under these conditions it is best to insert a pitot tube into the lead line from the well while operating under normal conditions. This involves the second division of the subject, namely, closed flow methods of measurement.

10 In closed flow measurements of gas where the volumes are small and the pressures are low, the ordinary positive gas meters are satisfactory, but for wholesale measurement some other type of meter must be adopted because of the prohibitive size of the positive type necessitated by large flows. The matter is further complicated

by the fact that usually when gas is bought or sold in large quantities it must be measured under varying pressures, necessitating the use of a correction factor to reduce the meter readings to quantities on a standard pressure base. There are many different types of proportional meters in use for this purpose, equipped with recording gages. The meter registration is then multiplied by a factor dependent upon the record of the gage in order to correct it to volumes based on the proper buying or selling pressure.

11 In addition to proportional meters there are a number of other devices in use which aim to attain a higher degree of accuracy, among which the most prominent are the venturi meter, the Thomas electric meter, the orifice, and the pitot tube.

12 The venturi meter is used principally for low pressure measurements, and while extremely accurate, is handicapped by possessing a complicated formula for the expression of volumes, which renders very tedious the computation of results from its records, and which demands a knowledge of the ratio of specific heats of the gas at constant pressure and constant volume, respectively. This formula is as follows:[1]

$$Q = 210{,}840\, A_2 \frac{T_s}{P_s} \sqrt{\frac{n}{(n-1)\, G}}\; \frac{P_1}{\sqrt{T_1}} \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \sqrt{\frac{1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{1}}}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}}} \quad \ldots\ldots [5]$$

wherein

Q = flow, cu. ft. per hr., on absolute pressure and temperature bases P_s and T_s
n = ratio of specific heats
G = specific gravity of gas
A_1 = area of venturi meter at entrance, in sq. ft.
A_2 = area of venturi meter at throat, in sq. ft.
P_1 = absolute pressure at entrance, lb. per sq. in.
P_2 = absolute pressure at throat, lb. per sq. in.
T_1 = absolute temperature at entrance, deg. fahr.
T_2 = absolute temperature at throat, deg. fahr.

13 For any particular meter and quality of gas, and assuming definite values of P_s, T_s, and T_1, equation [5] reduces to the simpler form

$$Q = KP_1R^{\alpha} \sqrt{\frac{1-R^{\beta}}{1-CR^{2\alpha}}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [6]$$

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[1] See Appendix No. 1.

where

$$K = 210{,}840\,A_2 \frac{T_s}{P_s} \sqrt{\frac{n}{n-1} \frac{1}{GT_1}}$$

$$C = \left(\frac{A_2}{A_1}\right)^2$$

$$R = \frac{P_2}{P_1}$$

$$\alpha = \frac{1}{n}$$

$$\beta = \frac{n-1}{n}$$

14 The Thomas electric meter [1] is an extremely accurate instrument for the measurement of gas and is comparatively simple in construction. It operates on the principle that the energy required to heat any quantity of a given gas through a definite temperature range is directly proportional to the weight of the gas heated, and can therefore be used as a measure of it. In practice, electrical energy is used to supply the heat and is automatically controlled in such manner as to cause the temperature increase of the flowing gas to remain fixed at a value of about 2 deg. fahr. The electrical energy required to do this is measured by means of wattmeters, the dials of which can be graduated to read direct in cubic feet of gas based on any desired standards of pressure and temperature. The method is independent of the actual temperature and pressure of the gas entering the meter, depending solely upon its specific heat at constant pressure. Since the records of the device are obtained through the medium of wattmeters, they can therefore be made to give either a rate reading of flow, an integrated or totalized reading, or both.

15 This meter has been subjected to very thorough tests in conjunction with the pitot tube and venturi meter [1] which have firmly established the accuracy and reliability of all three of the devices.

16 The Thomas meter requires a supply of electric energy equivalent to about 1 kw-hr. for each 75,000 cu. ft. of free gas. This fact greatly restricts its use in the measurement of natural gas, especially in the field where such power is not usually available.

17 The most practical devices known to the author for general use in the accurate measurement of gas are the pitot tube and the

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[1] Measurement of Gases, Carl C. Thomas, The Journal of the Franklin Institute, November 1911.

orifice, both of which are extremely simple in construction and are based on fundamentally correct principles. The formula expressing the flow of gas is identical for the two instruments for the range through which they are usually used.

18 The pitot tube was first described before the French Academy of Sciences in 1732 by its inventor, Pitot, who stated that it followed the principle of the law of falling bodies, namely $v^2 = 2\ gh$. For over a century after its discovery this wonderfully simple and accurate device was regarded merely as a scientific toy, until about 1850, when D'Arcy adapted it to closed pipe flows by the addition to the bent tube of a static pressure tube. It was then found that the difference in height of liquid in the two columns of a U-gage connected to the dynamic and static tips of the tube respectively, gave a true measure of the velocity, and followed Pitot's originally stated principle exactly, viz., $v^2 = 2\ gh$.

19 Prof. S. W. Robinson seems to have been the first to have applied the pitot tube to the measurement of gases. In 1873 he experimented with it in determining the velocity and form of jets of air from orifices, which led to the development of a commercial pitot tube for gas measurement.

20 If a pitot tube is inserted in a pipe conveying gas and a traverse made over the cross-section of the pipe, it will be found that the highest velocity exists at the center and the lowest at the walls. For this reason it has been customary for experimenters to determine the particular point in the cross-section of the pipe where the velocity corresponds to the mean velocity of the whole cross-section, and to keep the tube at that point for the determination of the quantity of flow as computed from the velocity readings. This point varies according to the condition of roughness of the interior surface of the pipe in which the tube is inserted, and accordingly it is necessary to determine it for each particular pipe used.

21 When a tube is to be used as a permanent gas meter, it is better, because of the fact just noted, to construct the pipe with extreme care, having the interior of uniform size and highly polished, and then to place the tip in the exact center of it. This necessitates the inclusion in the theoretical formula of a factor which in this paper is termed the "efficiency" of the tube, in order to reduce the flow, as indicated by the tube readings, to the value it would assume were the tube at the point corresponding to mean velocity instead of being at that of maximum velocity.

22 The tubes used by the author are all constructed in this man-

ner. The instrument as a whole is termed a pitot tube (comprising both the pipe and the bent tube inserted in it), and the bent tube portion of it is designated as the "tip." The pipe consists of 12 ft. of seamless drawn brass tubing with highly polished interior surface and gaged to accurate size. About 27 in. from the outflow end a brass saddle is sweated on, through which the tip projects into the tube, and which also contains the static pressure connection, which is merely a hole drilled through the saddle and tube flush with the inside wall. The tip is constructed in such manner as to be readily removable for cleaning, and yet exactly central in the tube when re-inserted.

23 A series of standard tubes of this type are maintained, against which all new tubes are calibrated. These standards, ranging in size from 2 in. to 6 in., were themselves calibrated by exhaustive tests against a large gas holder under different conditions of flow and pressure. The coefficients thus obtained were found to be constant throughout the complete range of water readings under which the tubes are used and also for all pressures.

24 The accuracy of these standards is attested by the paper by Professor Thomas already referred to, in which are given the results of a 45-day test at the Brave pump station of the People's Natural Gas Company of Pittsburgh, during which the Thomas electric meter was connected in series with pitot tubes that had been calibrated against the standards above referred to. A total of about 337,000,000 cu. ft. of gas was measured with a difference between the two meter records of 0.2 per cent.

25 The general formula[1] of the pitot tube and orifice is expressed by the equation

$$Q=218.44\ Ed^2\ \frac{T_s}{P_s}\sqrt{\frac{hP}{TG}}\ldots\ldots\ldots\ldots\ldots\ldots[7]$$

where

E = efficiency of the tube, expressed as a decimal
d = internal diameter of the tube, in.
T_s = absolute temperature of measurement base, deg. fahr.
P_s = absolute pressure of measurement base, lb. per sq. in.
G = specific gravity of gas — air = 1.0
P = absolute static pressure of flowing gas, lb. per sq. in.
T = absolute temperature of flowing gas, deg. fahr.
h = velocity head of flowing gas, in in. of water

[1] See Appendix No. 2.

Assuming standard conditions of storage temperature and pressure, namely, $P_s = 14.65$ lb. per sq. in. absolute, and $T_s = 60 + 460 = 520$ deg. absolute, equation [7] becomes

$$Q = 7753\ Ed^2 \sqrt{\frac{hP}{TG}} \qquad [8]$$

26 For any particular installation the gravity of the gas is once determined, and, with an assumed value for flowing temperature, is included with E and d to form a coefficient, making the formula read

$$Q = C\sqrt{hP} \qquad [9]$$

where

$$C = 218.44\ Ed^2 \frac{T_s}{P_s \sqrt{TG}} \qquad [10]$$

27 If the flowing temperature is other than the value assumed in making up the coefficient, the flow is corrected by multiplying by the factor $\sqrt{\frac{T_a}{T}}$, where T_a is the assumed flowing temperature.

28 The value of E is remarkably constant for tubes ranging in size from 2 in. to 5 in., as will be seen from the following tabulation, taken from the standard tubes already referred to:

Actual Diameter of tube, d	Efficiency, E
3.0625	0.8534
3.96875	0.8515
5.0469	0.8541
Mean	0.8530

29 Assuming the average value of E to be 0.8530, the coefficient of any pitot tube constructed as above described will be

$$C = 381.7d^2 \qquad [11]$$

based on a storage pressure of 14.65 lb. absolute, storage temperature of 60 deg. fahr., flowing temperature of 40 deg. fahr., and specific gravity of 0.600.

30 In using the pitot tube it is essential that a straight run of at least 30 ft. of pipe of the same diameter as the tube be installed ahead of it, in order to avoid errors due to eddies in the flow. It is also necessary to have absolutely no leaks in the pressure lines leading from the tube to the differential gage.

31 In practice the U-gage is modified by making one leg to consist of a cast-iron water pot having a cross-sectional area many times

greater than that of the glass tube comprising the other leg of the U. Thus the height of the water level in the pot does not change appreciably for a change in the glass tube from zero to full scale reading of about 28 in. The pot is provided with a gage glass to show the level of the water in it, to which the zero of the adjustable scale is set, thus necessitating the reading of but one leg of the U-gage, that comprised by the long gage glass.

32 The scales used in the tubes described are graduated to 28 in., and are used under all conditions of pressure. If the static pressure of the flowing gas is equal to that of the atmosphere, the full scale water reading corresponds to a velocity in the pipe of over 22,000 ft. per minute. As the static pressure increases, the velocity equivalent becomes smaller and at 100 lb. gage is about 8000 ft. per minute. Thus, the higher the pressure under which the tube operates, the greater will be the range of water pressure permissible to use.

33 It is not considered advisable to use the pitot tube for flows giving a differential pressure less than 2 in. of water, owing to the high percentage of error resulting from any personal error in reading the gage at such low readings. With a maximum scale reading of 28 in. this means that under atmospheric pressure conditions the ratio of maximum to minimum quantities of gas susceptible of accurate measurement by the pitot tube is about 3.75 to 1. Assuming the same permissible velocity (22,400 ft. per minute) under a pressure of 100 lb. gage, the maximum water scale reading could be increased to more than 200 in., giving the tube a range of 10 to 1. The author is now conducting experiments along this line, but as yet has not progressed far enough with the work to state authoritatively to what extent this range may be extended without affecting the constancy of the coefficient.

34 At present the usual method of obtaining the required range of capacity in a pitot tube measuring station is to install the tubes in batteries so that more or less of them may be cut in or out of service, as conditions demand, for obtaining accurate readings.

35 In accordance with Torricelli's theorem, the velocity of issue of a fluid from an orifice follows the law of falling bodies, and the formula deduced therefrom expressing the quantity of flow is identical with that of the pitot tube. The vena contracta of the jet, however, causes the coefficient of flow (termed "efficiency" herein) to have a lower value than the corresponding factor in the pitot tube formula. For differential pressures greater than about 4 in. of water

it is sensibly constant, and thus the orifice furnishes a simple means of gas measurement that compares favorably with the pitot tube.

36 Instead of a water column, requiring personal observation at regular stated intervals, a recording differential gage may be used, constructed with the working parts enclosed in a casing to which the static pressure is applied, while the dynamic pressure is applied to the interior of the gage. Such an instrument has been used with success in connection with an ordinary recording pressure gage for determining the static pressure, thus obviating the necessity of having an attendant constantly on duty to make observations.

37 The simplicity of the pitot tube and orifice, both in manipulation and computation, and the high degree of accuracy attainable with them, render them invaluable in the measurement of natural gas in large volumes and at high pressures. The tube has been in commercial use for many years, with highly satisfactory results. The orifice, although one of the oldest known measuring devices, has not until recently been used extensively in the measurement of gas under high pressure closed flow conditions. It is being rapidly developed, however, and bids fair to come into extensive use as a simple, accurate gas-measuring device.

38 The only disadvantage in the use of these two instruments is the necessity for the computation of results. Their largest field has consequently been in the measurement of sufficient quantities of gas to warrant the maintenance of stations with observers constantly in attendance. The adoption of the recording gages, however, has greatly extended their usefulness and has considerably simplified the problem of the accurate measurement of gas under pressure.

APPENDIX NO. 1

39 The general formula of the venturi meter is derived on the assumption that the change in state of the gas between entrance and throat of the meter takes place adiabatically. Hence the drop in potential energy, as evidenced by the respective pressures is equal to the increase in kinetic energy indicated by the respective velocities at entrance and throat.

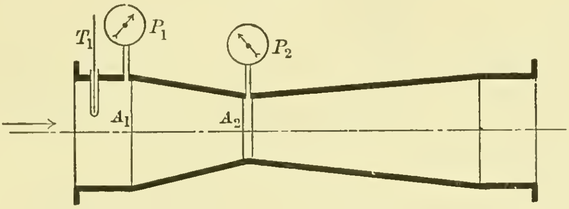

FIG. 1 DIAGRAM OF VENTURI METER

40 In Fig. 1, which diagrammatically represents a venturi meter, let

A_1 = area of cross-section at entrance, sq. ft.
A_2 = area of cross-section at throat, sq. ft.
P_1 = absolute pressure, lb. per sq. in. at entrance
P_2 = absolute pressure, lb. per sq. in. at throat
v_1 = absolute velocity at entrance, ft. per second
v_2 = absolute velocity at throat, ft. per second
W = weight of gas passing per second
δ_1 = density of gas at entrance, lb. per cu. ft.
δ_2 = density of gas at throat, lb. per cu. ft.
T_1 = absolute temperature of gas at entrance
T_2 = absolute temperature of gas at throat
T_s = absolute temperature of storage base, deg. fahr.
P_s = absolute pressure of storage base, lb. per. sq. in.
G = specific gravity of gas, air = 1.0
T_o = absolute temperature of 32 deg. fahr. = 492
P_c = absolute pressure of atmosphere, lb. per sq. in.
δ_w = weight of water per cu. ft. at 60 deg. fahr. = 62.37 lb.
δ_a = density of air at T_o and P_o
δ_{a1} = density of air at T_1 and P_1

Kinetic energy of gas at entrance $= \dfrac{Wv_1^2}{2g}$

Kinetic energy of gas at throat $= \dfrac{Wv_2^2}{2g}$

Increase in kinetic energy is

$$E_k = \frac{Wv_2^2}{2g} - \frac{Wv_1^2}{2g} = \frac{W}{2g}(v_2^2 - v_1^2) \quad \text{[12]}$$

41 The work done on one pound of gas in expanding adiabatically from P_1 to P_2 is

$$\frac{144\,(P_1V_1 - P_2V_2)\,n}{n-1}$$

and the change in potential energy in W pounds of gas per second is

$$E_p = \frac{144\,W\,(P_1V_1 - P_2V_2)\,n}{n-1} \quad \text{[13]}$$

42 Hence

$$\frac{W}{2g}(v_2^2 - v_1^2) = \frac{144\,Wn}{n-1}(P_1V_1 - P_2V_2) = \frac{144\,Wn}{n-1}\left(\frac{P_1}{\delta_1} - \frac{P_2}{\delta_2}\right)$$

$$v_2^2 - v_1^2 = \frac{288\,gn}{n-1}\left(\frac{P_1}{\delta_1} - \frac{P_2}{\delta_2}\right) \quad \text{[14]}$$

But $v_1A_1\delta_1 = v_2A_2\delta_2$ and $P_1\,\delta_2^{\,n} = P_2\delta_1^{\,n}$ or $\dfrac{\delta_2}{\delta_1} = \left(\dfrac{P_2}{P_1}\right)^{\frac{1}{n}}$. Hence

$$v_1 = v_2\,\frac{A_2\delta_2}{A_1\delta_1} = v_2\,\frac{A_2}{A_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

$$v_2^2 - v_1^2 = v_2^2 - v_2^2\left(\frac{A_2}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} = \frac{288\,gn}{n-1}\left(\frac{P_1}{\delta_1} - \frac{P_2}{\delta_2}\right)$$

$$v_2^2 = \frac{\dfrac{288\,gn}{n-1}\left(\dfrac{P_1}{\delta_1} - \dfrac{P_2}{\delta_2}\right)}{1 - \left(\dfrac{A_2}{A_1}\right)^2\left(\dfrac{P_2}{P_1}\right)^{\frac{2}{n}}} \quad \text{[15]}$$

43 But $\dfrac{P_1}{\delta_{a1}\,T_1} = \dfrac{P_o}{\delta_a\,T_o}$ and $\delta_{a1} = \delta_a\dfrac{P_1T_o}{P_oT_1}$

$$\delta_1 = G\delta_{a1} = G\delta_a\,\frac{P_1T_o}{P_oT_1}$$

Also

$$\frac{P_1}{\delta_1^{\,n}} = \frac{P_2}{\delta_2^{\,n}} \quad \therefore \quad \delta_2 = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\delta_1 = G\delta_a\,\frac{P_1T_o}{P_oT_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

Hence

$$\frac{P_1}{\delta_1} - \frac{P_2}{\delta_2} = \frac{P_1}{G\delta_a\,\dfrac{P_1T_o}{P_oT_1}} - \frac{P_2}{G\delta_a\,\dfrac{P_1T_o}{P_oT_1}\left(\dfrac{P_2}{P_1}\right)^{\frac{1}{n}}} = \frac{P_oT_1}{G\delta_a T_o}\left\{1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right\} \quad \text{[16]}$$

Substituting in [15], and extracting the square root

$$v_2 = \sqrt{\frac{\dfrac{288\,gn}{n-1}\,\dfrac{P_oT_1}{G\delta_aT_o}\left\{1-\left(\dfrac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right\}}{1-\left(\dfrac{A_2}{A_1}\right)^2\left(\dfrac{P_2}{P_1}\right)^{\frac{2}{n}}}} \quad\ldots\ldots\ldots\ldots\ldots\ldots[17]$$

44 The flow, in cubic feet per hour, based on P_s and T_s is

$$Q = 3600\ A_2v_2\,\frac{P_2}{P_s}\cdot\frac{T_s}{T_2}$$

But

$$\frac{P_2}{T_2} = \frac{P_1\delta_2}{T_1\delta_1} = \frac{P_1}{T_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

Hence

$$Q = 3600\ A_2\,\frac{T_s}{P_s}\,\frac{P_1}{T_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} v_2$$

$$Q = 210.840\ A_2\,\frac{T_s}{P_s}\sqrt{\frac{n}{(n-1)\,G}}\;\frac{P_1}{\sqrt{T_1}}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\sqrt{\frac{1-\left(\dfrac{P_2}{P_1}\right)^{\frac{n-1}{n}}}{1-\left(\dfrac{A_2}{A_1}\right)^2\left(\dfrac{P_2}{P_1}\right)^{\frac{2}{n}}}}\quad\ldots\ldots[18]$$

APPENDIX NO. 2

45 The general formula for the pitot tube and the orifice is derived from the law of falling bodies. Let

T = absolute temperature of flowing gas, deg. fahr.
P = absolute pressure of flowing gas, lb. per sq. in.
δ = weight per cu. ft. of flowing gas, at P and T
G = specific gravity of flowing gas, air = 1.0
v = actual velocity of flowing gas, ft. per second
h_f = height in ft. of homogeneous column of gas at P and T producing v
h = corresponding height of water column in in.
δ_w = weight per cu. ft. of water = 62.37 lb. at 60 deg. fahr.
P_s = absolute storage pressure base, lb. per sq. in.
T_s = absolute storage temperature base, deg. fahr.
δ_a = weight per cu. ft. air at 32 deg. and 14.7 lb. = 0.08073 lb.
d = actual inside diameter of pipe or orifice, in.
E = efficiency of pitot tube or orifice
Q = flow in cu. ft. per hour at P_s and T_s

Then

$$v = \sqrt{2\,g h_f} = \sqrt{2\,g\,\frac{h}{12}\,\frac{\delta_w}{\delta}}$$

But

$$\delta = \delta_a\,G\,\frac{P}{14.7}\,\frac{492}{T}$$

Hence

$$v = \sqrt{2\,g\,\frac{h}{12}\,\frac{\delta_w}{\delta_a}\,\frac{14.7}{492}\,\frac{T}{PG}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [19]$$

$$Q = 3600\,\frac{\pi\,d^2}{4 \times 144}\,v\,\frac{P}{P_s}\,\frac{T_s}{T}\,E$$

$$Q = 218.44\ Ed^2\,\frac{T_s}{P_s}\sqrt{\frac{hP}{TG}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [20]$$

THE CENTRIFUGAL BLOWER FOR HIGH PRESSURES

By Henry F. Schmidt

ABSTRACT OF PAPER

In this paper the author gives the essential elements of blower design in convenient form for reference. An equation is derived representing the work done in a centrifugal blower and the differences between this type and a piston compressor are discussed. There is a discussion of efficiencies and characteristics of impellers in series, of guide-vane and volute types of blowers and variations of pressure in the case of guide-vane blowers as the volume of discharge increases. The theory of diffusion vanes is examined in detail, and an equation for the loss of kinetic energy derived.

Design of volute blowers is taken up and discussion of the influence of the relation between efficiency curve and discharge rate, as well as between the blower characteristic and efficiency curve in this type of blower; an equation for the velocity conversion efficiency is derived, and the constancy of pressure in a blower having a free vorte mathematically demonstrated. The Westinghouse design of volute blower is used for illustration.

The latter part of the article is devoted to methods of testing and calculation of centrifugal blowers.

THE CENTRIFUGAL BLOWER FOR HIGH PRESSURES

By Henry F. Schmidt,[1] East Pittsburgh, Pa.

Non-Member

In presenting the following notes, the writer does not claim originality, but offers the material simply as a collection of the essential elements of blower design in convenient form for reference.

2 The centrifugal blower for high pressures has suffered through the same lack of knowledge in regard to its possibilities as that which affected the introduction of the centrifugal pump for high pressures and high shaft speeds. A general impression was created among engineers from early experiments that centrifugal pumps and blowers were adapted for handling only large volumes against low heads and with comparatively low efficiency. This, however, has been shown to be an incorrect statement in the case of centrifugal pumps and it is also untrue in respect to centrifugal blowers.

3 With proper methods of construction of the rotating parts and due regard to the theory involved, it is possible to build blowers for practically any capacity and pressure with efficiencies ranging from 65 per cent to 80 per cent or more, the higher efficiency, of course, being for the larger volumes handled.

4 Before an intelligent conception of the characteristics of centrifugal blowers can be obtained it is necessary to examine the theory, which is not only simple but extremely interesting.

5 The first problem to be considered is the pressure created within the impeller due to the centrifugal force acting on the volume of air between the blades of the fan. This can be done as follows:

[1] Westinghouse Machine Company.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

6 In Fig. 1, imagine a tube extending radially from the shaft to the circumference, and assume the area of this tube to be A sq. ft. If an infinitely thin lamina of this tube be taken, the weight of the air contained in it will be $\rho A dr$ where ρ is the density of the air at that point. If this lamina is taken at radius r, then the centrifugal force acting on this small mass of air will be

$$dF = \frac{\rho A dr \mu^2}{gr} = \frac{\rho A\ (2\pi)^2\ rN^2dr}{g}$$

in which N = revolutions per second. If the compression in the blades is adiabatic, the volume of a pound of air at any point is found from the relation $pv^\gamma = p_1v_1{}^\gamma$ or

$$v = v_1 \left(\frac{p_1}{p}\right)^{\frac{1}{\gamma}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [1]$$

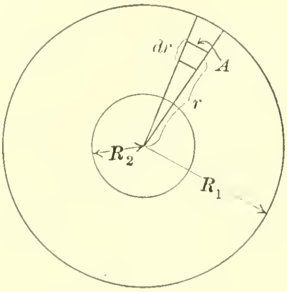

FIG. 1 PRESSURE WITHIN THE IMPELLER

But since the density is inversely proportional to the volume

$$\frac{1}{\rho} = \frac{1}{\rho_1}\left(\frac{p_1}{p}\right)^{\frac{1}{\gamma}} \quad \text{or} \quad \rho = \rho_1 \left(\frac{p_1}{p}\right)^{-\frac{1}{\gamma}}$$

7 Now also dF must equal the difference in pressure between the two sides of the lamina (dr) or $dF = Adp$. Hence, substituting the value of dF and ρ

$$dF = Adp = \frac{\rho_1 \left(\frac{p_1}{p}\right)^{-\frac{1}{\gamma}} A\ (2\pi)^2\ N^2rdr}{g}$$

Transposing $\rho_1 \left(\frac{p_1}{p}\right)^{-\frac{1}{\gamma}}$

$$\frac{1}{\rho} p_1{}^{\frac{1}{\gamma}} p^{-\frac{1}{\gamma}} dp = \frac{(2\pi)^2\ N^2rdr}{g}$$

Integrating between $\gamma=0$ and $\gamma=R$, and p_1 and p_2,

$$v_1\, p_1^{\frac{1}{\gamma}} \int_{p_2}^{p_1} p^{\frac{1}{\gamma}} dp = \frac{(2\pi)^2\, N^2}{g} \int_0^R r dr$$

which gives

$$v_1\, p_1^{\frac{1}{\gamma}} \frac{\gamma}{\gamma-1} \left[p_1^{\frac{\gamma-1}{\gamma}} - p_2^{\frac{\gamma-1}{\gamma}} \right] = \frac{(2\pi)^2\, N^2\, R^2}{2g} \ldots\ldots\ldots\ldots [2]$$

8 But the first term of this equation is the work done in adiabatic compression or expansion, and the second term equals $\frac{\mu_1^2}{2g}$, hence the work done in the impeller is equal to that required to accelerate the air from rest to the velocity of the blade tips. Since, however, if the radial velocity is very small, the final tangential velocity of the air is μ_1 it is evident that one-half of the total work done on the air by the blades will be expended in accelerating it to a velocity equal to that of the blade tips, and the other half in compressing it to a pressure p_2. Hence, unless the final velocity of the air is reduced by converting the kinetic into potential energy, the maximum efficiency of a centrifugal compressor, neglecting all losses, can be but 50 per cent. It is evident, therefore, that some efficient means must be provided to convert the kinetic energy represented by the final velocity into potential energy represented by an increase in pressure. This can be done in one of three ways:

a By means of diffusion vanes placed around the periphery of the impeller

b By the use of a volute of proper design

c By a diffusion tube or inverted nozzle attached to the outlet of the compressor

9 The first of these methods is probably the most efficient at approximately the designed capacity if properly constructed, but is inefficient at large and small discharge rates. The additional cost of construction is also one of its disadvantages.

10 Theoretically and practically, the second method is the most efficient over a wide range of loads and has, if well designed, an efficiency of 65 per cent to 90 per cent. It is, however, more costly to construct than the third method.

11 Diffusion tubes attached to the discharge, though the cheapest to construct, have only cheapness and compactness to recommend them and cannot be used on the intermediate stages of multi-stage compressors. The reason for the inefficiency of diffusion tubes is that serious eddies are formed in them due to the varying pressures

at different sections of the discharge opening. The latter results from the fact that the air farthest from the shaft is compressed the most and has a higher velocity, which causes eddies.

12 From an examination of equation [2] representing the increase of pressure or work in foot-pounds per pound of air in the impeller, it is evident that the most peculiar characteristic of the centrifugal blower is that for a fixed number of revolutions per minute, or tip speed of the impeller, the work done remains constant so long as the volume entering the blower remains constant. It will be noted that in this respect a centrifugal blower departs entirely from the piston compressor in that the volume discharged depends upon the pressure against which the blower is working, whereas in the piston blower for a fixed number of revolutions per minute the volume remains practically constant, the pressure of discharge varying according to the resistance placed in the discharge.

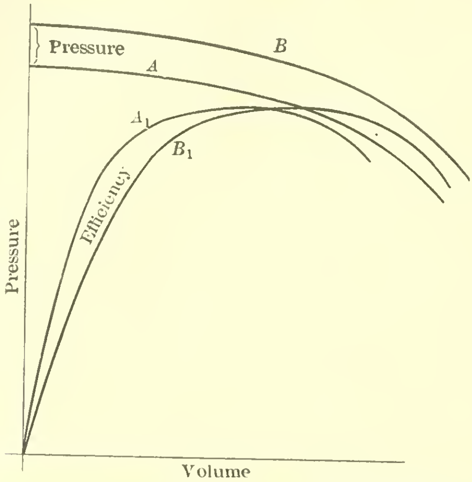

FIG. 2 PRESSURE AND EFFICIENCY CURVES, CENTRIFUGAL BLOWER

13 It will be noted that the work done in a centrifugal blower varies directly as the square of the tip speed, or if the diameter of the impellers is fixed, directly as the number of revolutions per minute and the capacity almost directly as the number of revolutions per minute. The result is that the pressure and efficiency curves for a change in the number of revolutions per minute vary somewhat from the manner shown diagrammatically in Fig. 2. The point to

be noted particularly in regard to Fig. 2 is that the maximum efficiency obtained is practically independent of the number of revolutions per minute, since all the losses in the blower vary directly as the cube of the number of revolutions per minute, as does also the total work done per pound of air. Consequently, the losses bear the same relation to the total input at a small number of revolutions or at a low pressure discharged against as they do for a higher number of revolutions and higher pressure. For this reason the efficiency curve characteristic remains the same for changes of speed, but the point of maximum efficiency shifts to a smaller volume with decrease in the number of revolutions per minute.

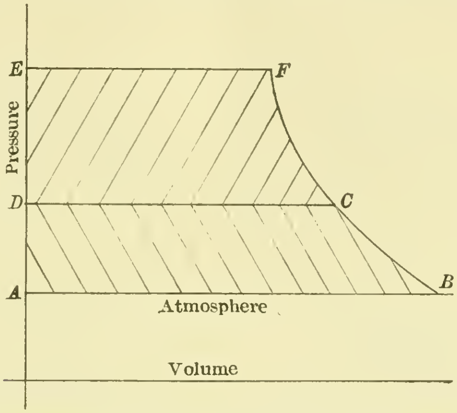

Fig. 3 Diagram of Work done by Two Impellers in Series

14 As the work done per pound of air is independent of everything except the tip speed and the volume of the air entering the impeller, it is evident that the air discharged from one impeller may be supplied to a second impeller, which will add the same number of foot-pounds of work per pound of air to it, thus increasing the pressure by adding that created by the first impeller to that created by the second. The effect of this will be seen by examining Fig. 3 in which the area *BADC* represents the work done by an impeller compressing from atmospheric pressure to a pressure *D*. Where the air is discharged into a second impeller doing the same amount of work or the area as represented by the area *CDEF*, which is equal to the area *BADC*, it should be noted that, owing to the decrease of volume with increase of pressure, the actual difference of pressure created in the second impeller is greater than that in the first, so

that high pressures can be attained with a comparatively small number of impellers or stages in series. The efficiency of such a combination of impellers placed in series is exactly the same as though each impeller were an independent unit, the total efficiency being the same as that of the individual stages.

15 Figs. 4, 5, 6 and 7 show diagrammatically the construction and characteristics respectively of the guide vane and volute types of blower. As will be noted by examining Fig. 4, the guide vanes consist of a number of curved vanes surrounding the impeller or rotor and pointing in the direction of rotation. Since the air discharged from the impeller enters the diffusion vanes at the narrowest portion at a high velocity and leaves at the point of greatest cross-section with a greatly reduced velocity, the difference in kinetic energy between the point of entrance and exit is partly utilized in increasing the pressure of the air and partly in eddies and friction, the latter part reappearing in the form of heat. The characteristic of this type of blower is seen in Fig. 6, which shows that the pressure at first increases as the volume of discharge increases until it reaches a maximum, after which the pressure again falls off, and it will also be noted that at constant discharge (and constant revolutions per minute) the pressure does not remain constant but is continually fluctuating between limits, as shown by the shaded lines, which usually represent about 10 to 15 per cent of the total pressure created. The reason for this will be understood by examining Fig. 8.

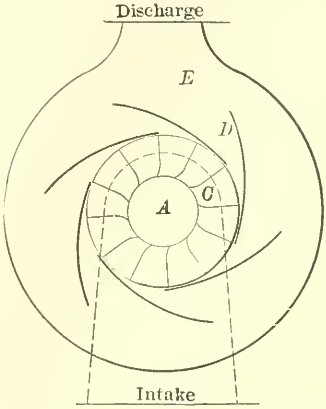

Fig. 4 Guide Vane Blower

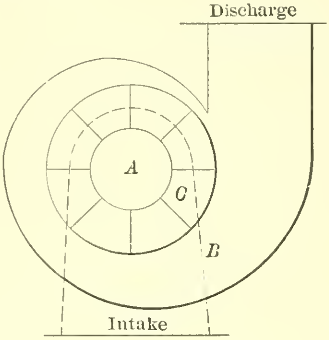

Fig. 5 Volute Blower

16 For the sake of illustration, instead of assuming the air to

leave the impeller, imagine it to issue from a nozzle through which the air is supplied at a pressure P_1 and expands to a pressure P_2 such that the velocity at the mouth of the nozzle is exactly equal to the velocity with which it leaves the tips of the impeller blades and enters the guide vanes. Further, in place of the guide vanes, assume a straight diffusion tube discharging into a receiver R. The pressure in it is controlled by means of a valve or other device. Hence, from the assumptions made, the diffusion tube when supplied with air as stated is an exact equivalent of the guide vanes in the blower shown in Fig. 4. Now, since the area of the diffusion tube increases, the velocity of the air must decrease, with the result, as previously mentioned, that the kinetic energy is partly converted into pressure, as indicated by the diagram beneath the diffusion tube. The line A_1 is the final pressure resulting from the velocity attained by the air in expanding in nozzle from pressure P_1 at A to the back pressure P_2, and the line B_2 represents the final pressure of the air when expanded from the initial pressure B to the final pressure P_2. Now it will be evident from the method in which this velocity conversion takes place that there can be but one pressure of equilibrium for any given velocity with which the air enters the diffuser. Furthermore, if the pressure P_3 in the receiver R is built up to the point where air can no longer enter the inlet of the diffuser, the pressure in the receiver must break down, for with no air entering the diffuser the velocity becomes zero, and velocity conversion is no longer possible. The result of this condition is that in a centrifugal blower fitted with guide vanes, under certain conditions the pressure may break down to the point where the only pressure available at the discharge is that due to the impeller alone, and even with normal rates of discharge there is also a tendency for the diffuser vanes to build up a

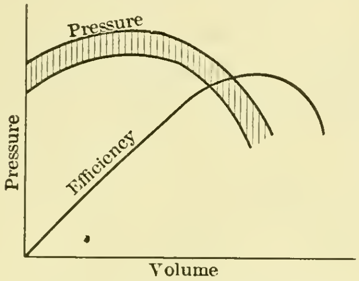

Fig. 6 Guide Vane Blower Characteristics

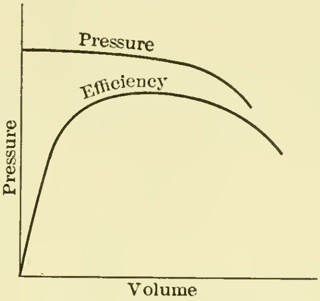

Fig. 7 Volute Blower Characteristics

pressure higher than that in the receiver R, resulting in a constant building up and breaking down of the pressure, as shown in Fig. 9.

17 The rapidity with which these variations occur depends upon the volume of the receiver space and pipe line into which the blower is discharging. The smaller the receiver volume the more rapid the fluctuations and the smaller their extent. With very large receiver volumes and especially with a number of such blowers operating in parallel, fluctuations may occur in which the pressure breaks down to that of the impeller alone. To obviate this, it is customary to throttle such blowers at the intake, so as always to operate them below the lower pressure line in Fig. 6, thus avoiding the fluctuations of pressure.

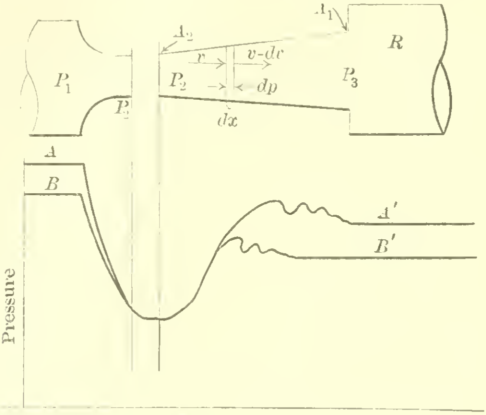

Fig. 8 Pressure Variations, Nozzle and Diffusion Tube

18 The fact that no pressure is created by the diffuser tube when no air is being delivered is the reason for the increase of pressure with increase of volume delivered up to a certain point beyond which the diffusion tubes are no longer of sufficient exit areas to reduce the final velocity to the necessary point, which in addition to the increased friction at the higher velocities causes a dropping off of the pressure line, as shown in Fig. 6. The result of this is that the efficiency curve rises very slowly at first and becomes a maximum over a comparatively very short range, when it immediately begins to drop off very rapidly.

19 The above actions will be better understood by examining the theory of diffusion vanes, in connection with Fig. 8, as follows:

20 Let the area of the smaller or entrance end of the diffuser be A_2 and the exit area be A_1; also assume that no heat is received by or given up by the diffusion tube, in which case the compression will be adiabatic (though in practice not necessarily isentropic) and the relation between pressure and volume at any point will be determined by the equation

$$pv^\gamma = pv_2^\gamma \quad \text{or} \quad v = \left(\frac{p_2}{p}\right)^{\frac{1}{\gamma}} v_2$$

21 In Fig. 8 consider an infinitely thin section of the tube dx and assume that the pressure at the point the section is taken is p, and the velocity on one side of the section is V and that the velocity on the other side is $V - dV$. Also assume that the mass of air within the section considered meets with such a resistance as will be produced by the force $dF = Adp$, A being the area of the tube at the section taken.

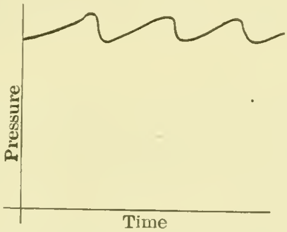

FIG. 9 PRESSURE VARIATIONS, GUIDE VANE BLOWER

22 Now the mass of the air in the section will obviously be $dw = PAdx$. But force is equal to mass times acceleration, hence,

$$dF = \frac{\rho A dx}{g} \frac{v - dv}{dt} = \frac{\rho A dx}{g} \frac{d^2x}{dt^2}$$

and also

$$dF = Adp$$

Hence equating these two expressions for dF

$$Adp = \frac{\rho A dx}{g} \frac{d^2x}{dt^2}$$

but $\rho = \frac{1}{v}$

hence

$$Adp = \frac{Adx}{v_2 \left(\frac{p_2}{p}\right)^{\frac{1}{\gamma}} g} \frac{d^2x}{dt^2}$$

or

$$gv_2\ p_2^{\frac{1}{\gamma}}\ \frac{dp}{p^{\frac{1}{\gamma}}} = \frac{d^2x}{dt^2}\ dx$$

which may be integrated between V_1 and V_2 and p_2 and p_1, and will give

$$2g\ \frac{\gamma}{\gamma-1}\ RT_2\left[\left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right] = -\left[V_1^2-V_2^2\right] \ldots\ldots\ldots[3]$$

that is, since kinetic energy has disappeared into potential energy the sign is negative. The most interesting part of this equation is that an examination shows that the kinetic energy lost in changing from a velocity V_1 to velocity V_2 is exactly the same as that given up in adiabatic expansion from the pressure P_1 to the pressure P_2. In other words, neglecting friction eddies and shock, for some fixed relation between the velocity of approach and final pressure, as indicated by the formula, the theoretical velocity conversion efficiency of a diffusion tube is unity, and constant delivery is likewise theoretically possible.

23 Returning now to the volute blower shown in Fig. 5, which is not fitted with guide vanes, an entirely different characteristic is obtained, the theoretical characteristic of this blower being a perfectly straight line, that is, constant pressure for any delivery, neglecting, of course, the energy put into the air in order to move it through the blower. In this type of blower, as there is no velocity conversion in the volute, the pressure remains the same at all rates of delivery except for the increased frictional resistance with higher velocities and larger discharge rates, which results in a gradual decrease of the pressure as shown in Fig. 7.

24 However, the most remarkable point in this design of blower is the rapidity with which the efficiency curve rises with increase of discharge rate and the wide range over which it remains practically constant. Though not plotted in scale, the pressure characteristics and efficiency characteristics shown in Fig. 7 are relatively exact, as found in the two types of blower, the maximum efficiency obtainable in both the volute and guide vane being for both purposes identical, provided, of course, that both are properly designed; but the point which should be noted particularly is that in commercial operation the value of a blower operating on fluctuating load depends not upon the maximum efficiency obtained, but upon the area under the efficiency curve, since this is a true measure of the average efficiency over a range of load.

25 The results from two blowers on actual test are given in Fig. 10, which clearly show the difference in the characteristics and the area under the two efficiency curves.

26 In order to find the pressure created in a free vortex, assume a tube extending from R_1 to R_3, Fig. 11, of area A and take a lamina of this tube of thickness dr at any radius r. Then the centrifugal force acting on this small mass of air will be

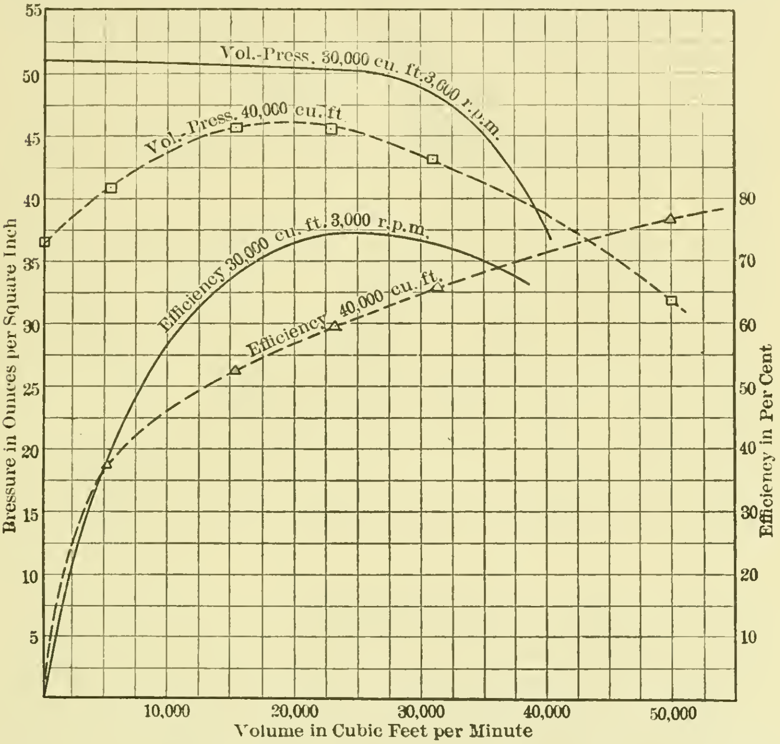

FIG. 10 CHARACTERISTICS OF TWO BLOWERS UNDER DIFFERENT EFFICIENCY CURVES

$$dF = A\,dp = \frac{\rho A\,dr\mu^2}{gr} \quad \text{or} \quad v\,dp = \frac{dr\mu^2}{gr}$$

But μ is inversely proportional to the radius, hence the velocity μ at any point will be $\frac{\mu_1 R_1}{r}$ where μ_1 is the tip velocity of impeller. Substituting the value of μ,

$$v\,dp = (\mu_1\ R_1)^2\frac{dr}{gr^3}$$

and integrating between the limits p_2 and p_3 and $r = R_1$, and $r = \infty$,

$$\frac{\gamma}{\gamma - 1}\, 2g\, p_2 v_2 \left[\left(\frac{p_3}{p_2}\right)^{\frac{\gamma - 1}{\gamma}} - 1\right] = -\frac{\mu_1^2 R_1^2}{\infty^2} + \frac{\mu_1^2 R_1^2}{R_1^2} = \mu_1^2$$

which shows that for a 100 per cent conversion efficiency the volute must be of infinite diameter. For any given radius of volute R_3 the work done by the velocity conversion in foot-pounds will be

$$W = \frac{\mu_1^2}{2g}\left[1 - \left(\frac{R_1}{R_3}\right)^2\right]$$

or the velocity conversion efficiency is

$$E = 1 - \left(\frac{R_1}{R_3}\right)^2$$

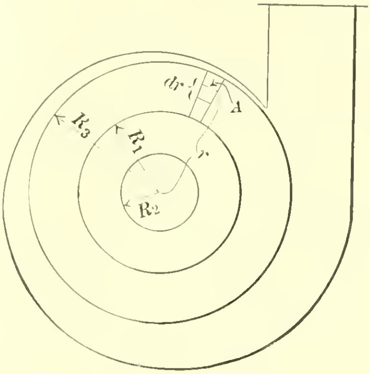

Fig. 11 Pressure Diagram in a Volute Blower with a Free Vortex

27 Now the reason for the statement that the pressure remains practically constant in a blower having a free vortex will be evident from the preceding mathematical demonstration, since it follows that as the pressure created in the vortex is due to the centrifugal force acting on the fluid in the vortex, neglecting friction and eddies, the pressure created must be independent of the volume delivered. This must be true for the further reason that the volume delivered does not enter into the mathematical analysis as a function of the work done, whereas, in the mathematical deduction of the equation of flow (or work done) in a diffusion tube, the volume enters the equation in the expression $(V_1^2 - V_2^2)$.

28 As previously stated, neglecting friction, the energy transformation taking place in a free vortex remains constant regardless of the volume delivered, even though the delivery is reduced to zero,

and likewise the efficiency is for the same reasons constant, independent of the volume delivered. This explains the much higher efficiency of the free vortex blower at partial rating than is obtained in a blower fitted with guide vanes. Further, when the element of friction is considered, the free vortex blower has a considerable advantage over the diffusion vane blower because the surface exposed to friction is very much smaller.

29 The preceding remarks are not based on theory alone, since the writer has made a large number of tests with pitot tubes and thus verified the above statements. These tests showed that, when properly constructed, theory and practice agree to a surprising extent.

30 In a discussion before the Society some time ago the design of the impellers used by the Westinghouse Machine Company was criticized as inefficient because the blades are radial and have no curvature at their inlet ends for picking up the incoming air without shock losses.

31 It has been shown that if the blades of an impeller extend all the way into the center of the shaft, the work done by compression in the rotor is just one-half of the total work done on the air, the other half being represented by the kinetic energy in the air leaving the impeller with the velocity of the blade tips, that is, energy represented by $\frac{\mu_1^2}{2g}$ per lb.

32 In actual practice the inner diameter of the blades, or "eyes," is usually about one-half the diameter of the impeller, or say just one-half, then the velocity of the entrance edges of the blades is one-half the tip velocity, that is, $\mu_2 = \frac{1}{2}\mu_1$. But the total energy put into the air is $2\left(\frac{\mu_1^2}{2g}\right)$ per lb., and since the velocity at the blade entrance is $\frac{1}{2}\mu_1$ the maximum loss from shock, if the air hits the blades at right angles, as in Fig. 12, would be $\frac{(\frac{1}{2}\mu_1)^2}{2g}$ and the percentage loss of the total energy put into the blower would consequently be

$$100\frac{\frac{(\frac{1}{2}\mu_1)^2}{2g}}{2\left(\frac{\mu_1^2}{2g}\right)} = 12.5$$

33 The preceding is on the assumption that all the velocity of

the air relative to the blade is destroyed, which, however, is far from the truth.

34 Since it is necessary for the air to have a radial velocity in order to pass through the impeller (usually the radial velocity of the air is about one-quarter of the tip velocity, or one-half of the velocity of the entrance of the blades), instead of striking the blades at right angles, it enters the blades about as shown in Fig. 13 at an angle of 45 deg. Hence, if a total loss by shock of the velocity component perpendicular to the blade is still assumed, the loss at the entrance is reduced from 0.125 of the total energy to about 0.06 of the total energy put into it. However, it is known that when air strikes a surface at an angle, there is a tendency to build up a bank of air, as shown in Fig. 14, and since the flow is continuous, all the energy is not lost by shock. Hence, it is very conservative to estimate the loss at the entrance with radial blades as not exceeding 1 or 2 per cent. This is also verified by the tests which have been made, as it was impossible to find the loss at the entrance because of its being so small compared with the total energy put into the air.

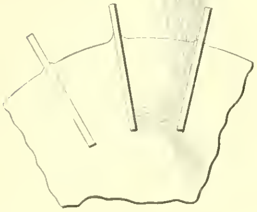
FIG. 12 STREAM LINES, AIR HITTING BLADES AT RIGHT ANGLES

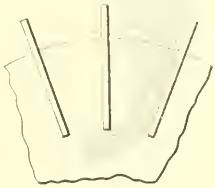
FIG. 13 STREAM LINES, AIR HITTING BLADES AT 45 DEG.

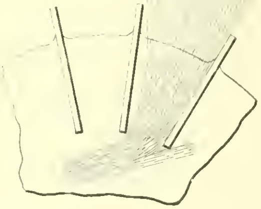
FIG. 14 BUILDING UP AN AIR-BANK, AIR HITTING BLADES AT AN ANGLE

35 Another reason why the loss is practically nothing in the Westinghouse design is that the inlet of the blowers is made in the form of a volute and forced vortex as shown in Fig. 15, thus causing the air to rotate in the same direction as the impeller before entering the impeller, and the proportions of this volute are so made that at normal load the air enters the impeller blades tangentially without any shock loss whatever.

36 Since centrifugal blowers for all except the very lowest pressures must run at comparatively high tip speeds, it becomes imperative that the rotor construction shall be of the very best design

possible, and all the stresses in the blades must be radial. Hence, a straight radial blade is the only one which can be used except at the very lowest speeds.

37 No riveted or built up construction can be satisfactory and only an impeller of the double-flow type milled out of the solid should be tolerated. Milling the rotor in halves and riveting them together is also a very poor construction, and has only cheapness to recommend it. The disk should be of such section as to give uniform stress throughout, and there should be no hub or hole for the shaft.

THE TESTING AND CALCULATION OF CENTRIFUGAL BLOWERS

38 The following notes give the readings to be taken, together with the formula used for working out the results of tests, as well as the method of correcting to standard or any selected contract conditions.

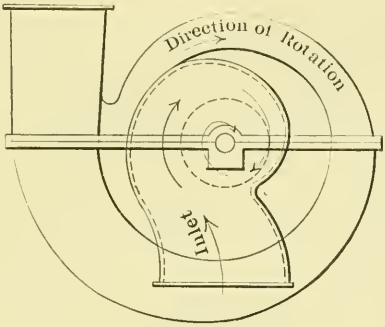

Fig. 15 Inlet of Blower, Westinghouse Design

39 The data which must be obtained for a complete test of a centrifugal blower comprise

- *a* brake horsepower input to the blower shaft
- *b* inlet temperature
- *c* inlet pressure of the air entering the blower
- *d* barometer reading
- *e* static pressure in the blower discharge, and temperature in the blower discharge
- *f* total head blown against, i. e., static head plus velocity by means of a pitot tube placed above the discharge nozzle, and flush with its outlet.

40 In obtaining the correct inlet temperature to the blower, the

temperature should be taken by means of a number of thermometers placed in different portions of the intake duct, so as to obtain a true average temperature, and precautions should be taken to see that no air, other than that passing the thermometers, is entering the blower intake, such as, for instance, air drawn from the basement below the engine-room floor.

41 The pressure in the blower discharge should be taken at a point as shown in Fig. 18, in which the distance h should be at least four or five times the outlet diameter of the nozzle, in order to avoid a reduction of pressure, which occurs near the nozzle.

42 The temperature also should be taken at a corresponding point by inserting one or more thermometers into the discharge.

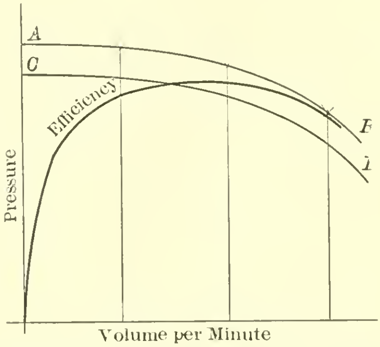

FIG. 16 PRESSURE AND EFFICIENCY CURVES, CENTRIFUGAL BLOWER

This can be most conveniently done by sticking the thermometers through rubber inserted in the discharge pipe.

43 The method of obtaining the total pressure, or static plus velocity head, will be seen from Fig. 18. Either mercury U-tubes or water manometers may be used for measuring the discharge pressure. In either case, the difference between the two sides of the U, i. e., the head in inches of mercury, should be corrected for the temperature of the U-tube.

44 The test nozzles should be made about as shown in Fig. 18; the radius or curvature of the approach to the nozzle should be equal to d, in which case the coefficient of the nozzle is for all practical purposes 100 per cent, or equal to the theoretical discharge. It has been generally agreed that a nozzle coefficient of 98.5 is, if anything,

probably 1 per cent low, and hence has been adopted where it is not convenient to calibrate the nozzle used.

METHOD OF CALCULATION

45 In the following calculation let

P_2 = absolute inlet pressure to the blower in inches of mercury (i. e., barometer reading)

P_1 = total head (static plus velocity) in inches of mercury, plus the barometer reading in inches of mercury

T_2 = absolute inlet temperature in deg. fahr.

T_1' = temperature in the discharge in deg. fahr. absolute

T_1 = theoretical temperature in the piping after adiabatic compression

T_2' = temperature at the mouth of the discharge nozzle.

46 The work done in foot-pounds per pound of air, in compressing from P_1 to P_2 will be expressed by the formula

$$W=\frac{\gamma}{\gamma-1}RT_2\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]\ldots\ldots\ldots\ldots\ldots[4]$$

in which R has a value of 1.4065, and the work done per pound of air in expanding from the pressure P_1 to atmospheric pressure in the nozzle will be

$$W=\frac{\gamma}{\gamma-1}RT_2'\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]\ldots\ldots\ldots\ldots\ldots[5]$$

in which

$$T_2'=\frac{T_1'}{\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}}\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots[6]$$

Hence, the velocity of the air in feet per second will be

$$V=8.025\sqrt{\frac{\gamma}{\gamma-1}RT_2'\left[\left(\frac{P_1}{P_2}\right)^{\frac{y-1}{\gamma}}-1\right]}\ldots\ldots\ldots\ldots[7]$$

Accordingly, the volume of discharge in cubic feet per second will be

$$\text{Volume per second} = AV\ldots\ldots\ldots\ldots\ldots\ldots[8]$$

in which A is the area of the mouth of the nozzle in square feet times the nozzle coefficient, which, as above stated, for nozzles properly made, is practically unity, or say 99 per cent. The total work done in foot-pounds per minute will be equation [4] × equation [8] × 60 ÷ specific volume of 1 lb. of air at the barometric pressure and temperature T_2, which may be expressed by the equation

$$W_{min}=\frac{\frac{\gamma}{\gamma-1}RT_2\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]\times 8.025\sqrt{\frac{\gamma}{\gamma-1}RT_2'\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]}\times A\times 60}{\frac{RT_2'}{P_2'}}\quad[9]$$

in which P_2' is the barometric pressure expressed in pounds per square foot absolute. Hence from equation [9] the theoretical air horsepower required to compress the column handled from the pressure P_2 to the pressure P_1 will be equation [9] divided by 33,000, or air h.p.

$$=\frac{\frac{\gamma}{\gamma-1}RT_2\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]\times 8.025\sqrt{\frac{\gamma}{\gamma-1}RT_2'\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]}\times A\times 60}{\frac{RT_2'}{P_2'}\times 33{,}000}\quad\ldots[10]$$

47 The efficiency of the blower is equal to the theoretical horsepower required to compress the given volume of air handled divided by the brake horsepower input at the shaft; hence, the efficiency of the blower is expressed by the equation

$$E=\frac{\text{air horsepower}}{\text{motor input}\times\text{motor efficiency}}\quad\ldots\ldots\ldots\ldots[11]$$

and the total overall combined efficiency will be

$$E_e=\frac{\text{air horsepower}}{\text{motor input}}\quad\ldots\ldots\ldots\ldots\ldots\ldots[12]$$

in each of which the motor input has been reduced to horsepower.

48 The blower efficiency may also be found by the temperature-pressure method in any blower which is not water-jacketed, the procedure in this case being to calculate the theoretical temperature rise from the formula

$$T_1-T_2=T_2\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]\quad\ldots\ldots\ldots\ldots\ldots[13]$$

dividing it by the actual temperature rise found by subtracting the inlet temperature from the temperature in the blower discharge, which gives rise to the equation

$$E'=\frac{T_2\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]}{T_1'-T_2}\quad\ldots\ldots\ldots\ldots\ldots[14]$$

49 Following is the derivation and use of formula [14].

To Prove: the efficiency of a blower is measured by the formula

$$E' = \frac{T_1 - T_2}{T_1' - T_2}$$

where

T_1 = theoretical final temperature of the air calculated from the adiabatic relation between pressure and temperature

T_2 = initial temperature of entering air

$T_1{}^1$ = final observed temperature in actual compressor.

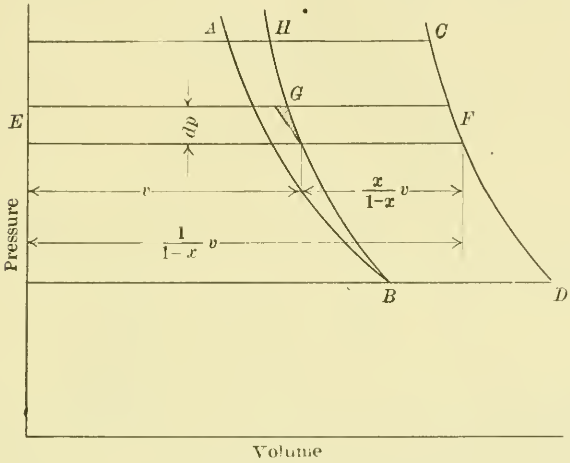

Fig. 17 Work Diagram, Centrifugal Blower

ANALYSIS

50 Any losses of energy which occur in a blower must reappear in some other form, namely, as heat. Hence, the various losses of whatever nature occurring in a blower may be considered equal to x. The losses include friction, eddies, leaking, etc. Referring to Fig. 17, let AB be the theoretical adiabatic compression curve and HB the actual compression line. Let the section EG be an infinitely small lamina of height dp. Since the work of vdp must be $\frac{1}{1-x}vdp$ extending the area EG to a point F on an imaginary compression curve CD. Note that vdp is the work done on the air as shown by the area of the actual compression curve and not under the adiabatic. Since, from the fundamental assumption that energy lost is converted into heat, the first law of thermodynamics

$$dQ = \frac{1}{\gamma - 1} vdp + \frac{\gamma}{\gamma - 1} pdv \qquad \ldots\ldots\ldots\ldots [15]$$

may be used. Substituting for dQ its value, the area

$$GF = \frac{x}{1-x} vdp$$

equation [15] becomes

$$\frac{x}{1-x} vdp = \frac{1}{\gamma - 1} vdp + \frac{\gamma}{\gamma - 1} pdv \qquad \ldots\ldots\ldots\ldots [16]$$

Integrating equation [16] between p_2 and p_1 and v_2 and v_1

$$v_1 = v_2 \left(\frac{P_2}{P_1}\right)^{\frac{1 + \frac{x}{1-x} - \frac{x}{1-x}\gamma}{\gamma}} \qquad \ldots\ldots\ldots\ldots [17]$$

$$\beta = \frac{1 + \frac{x}{1-x} - \frac{x}{1-x}\gamma}{\gamma} \qquad \ldots\ldots\ldots\ldots [18]$$

Substituting the value of v found from equation [17] in the equation $dW = vdp$ and integrating,

$$W = \frac{RT_2}{(1-x)\ (1-\beta)} \left[\left(\frac{P_1}{P_2}\right)^{(1-\beta)} - 1\right] \qquad \ldots\ldots\ldots\ldots [19]$$

51 Equation [19] gives the work done in foot-pounds per pound of air compressed from P_2 to P_1 including all losses of whatever nature occurring in the blower. It will therefore be evident that the efficiency of the blower will be equal to the theoretical work required to compress one pound of air from P_2 to P_1 divided by the actual work required for the same compression. Hence

$$E' = \frac{RT_2 \left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]^{\frac{\gamma}{\gamma-1}}}{RT_2 \left[\left(\frac{P_1}{P_2}\right)^{1-\beta} - 1\right] \frac{1}{(1-x)\ (1-\beta)}} \qquad \ldots\ldots\ldots [20]$$

but

$$\frac{1}{(1-x)\ (1-\beta)} = \frac{\gamma}{\gamma - 1}$$

also

$$T'_1 - T_2 = T_2 \left[\left(\frac{P_1}{P_2}\right)^{1-\beta} - 1\right]$$

hence the expression for the efficiency reduces to

$$E' = \frac{\frac{\gamma}{\gamma-1} RT_2 \left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} \quad 1\right]}{\frac{\gamma}{\gamma-1} RT_2 \left[\left(\frac{P_1}{P_2}\right)^{1-\beta} - 1\right]} = \frac{T_1 - T_2}{T'_1 - T_2} \qquad [21]$$

52 At this point it will be well to take up the old formulae used in a number of textbooks dealing with the measurement of air, viz.,

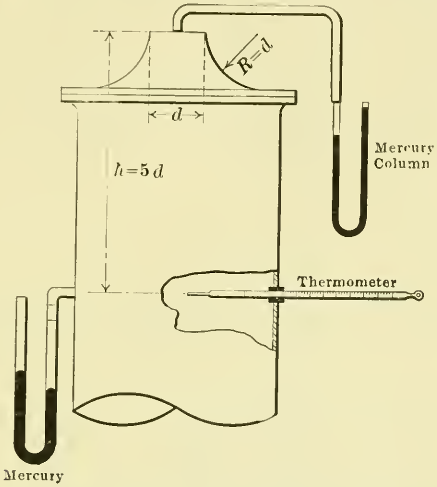

FIG. 18 TEST ARRANGEMENT FOR OBTAINING TOTAL PRESSURE

$$V = \sqrt{2gh} = 8.025\sqrt{h} \qquad [22]$$

in which h is the head through which the medium flows. In the case of air, or any gaseous medium, h being taken as

$$h = \frac{P_1 D \times 13.596}{12\rho} \qquad [23]$$

in which

P_1 = static, plus velocity head, in inches of mercury
D = weight of 1 cu. ft. of water at T'_2
ρ = weight of 1 cu. ft. of air at temperature T'_2 and barometric pressure.

53 Equation [23] is substantially correct for the measurement of velocity for comparatively low pressures, but gives a velocity slightly too high when employed for pressures of 2 or 3 lb. or more, in which case the velocity, as previously pointed out, should be

calculated from the formula [7] the error arising from the use of formula [22] being shown in Fig. 19. The work done is represented by $2gh$, being the area $BCDE$, whereas the actual area is that under the adiabatic curve represented by the curve $BCDA$.

54 The method of correcting for temperature at inlet to blower, and for changes of barometer is as follows:

55 The fundamental characteristic of the centrifugal blower is

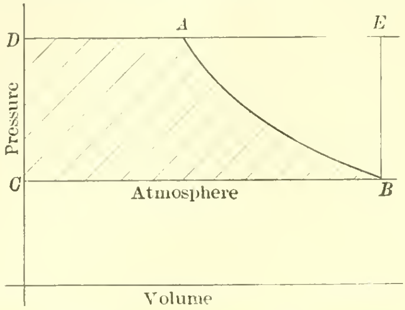

Fig 19 Work Diagram, showing difference between Textbook and Corrected Formulae

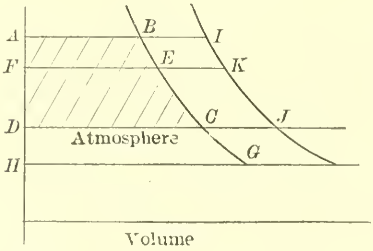

Fig. 20 Indicator Diagrams of Centrifugal Blower for Different Conditions of Inlet Temperature and Pressure

that for any given actual volume entering the blower inlet, the blower at a given speed will take a certain number of foot-pounds of work per pound of air, regardless of the intake temperature or pressure, and the result of this is that the ordinary methods of correcting for temperatures and barometer as applied to blowing engines, does not apply to centrifugal blowers. This will be seen from Fig. 20, which shows an indicator diagram of the centrifugal blower for different conditions of inlet temperature and pressure. Let the area $CDAB$ be the compression over the standard temperature and barometer, to some given pressure. This area will represent a fixed number of foot-pounds of work per pound of air. It is evident, therefore, that if the barometer readings were the same but the intake temperature higher, i. e., so that the initial volume were to point *J* instead of *C*, it is evident that the pressure to which the blower could compress would no longer be *A*, but some pressure *F*, such that the area *JDFK* would equal the area *CDAB*.

56 Consequently, the correction for change of temperature from *C* to *J* would not be applied as affecting the actual volume entering the blower, but as a correction to the pressure. In the same way, supposing that the barometer were low and the atmospheric line represented by *HG* at standard temperature, the area *GHFE* must again equal the area *CDAB*, and consequently the pressure would be lower than it would be with the standard barometer. This is the correction for both inlet temperature and barometer, to the pressure of compression and not to the inlet volume. As previously stated, the work done in foot-pounds per pound of air remains constant; hence, having found the work in foot-pounds per pound of air over the actual test conditions, it is possible to substitute in equation [4], the value of the standard inlet temperature in deg. fahr. absolute, and a standard barometer reading, and in this way to calculate the final pressure in the discharge for the standard conditions.

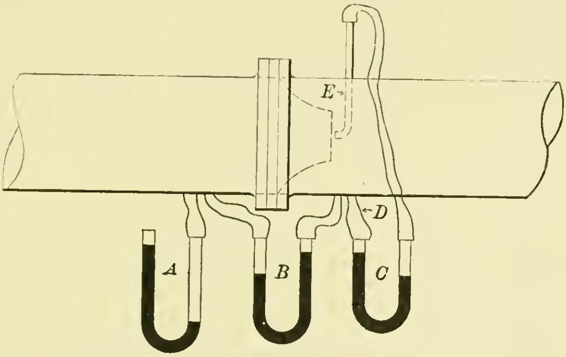

Fig. 21 Arrangement of Test for determining Flow in Pipe

57 If P_2' and T_2' are the absolute pressure and temperature at the inlet to which the test results are to be corrected, it follows that the corrected discharge pressure P_1' will be given by the equation,

$$P_1' = P_2' \left\{ \frac{T_2}{T_2'} \left[\left(\frac{P_1}{P_2} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] + 1 \right\}^{\frac{\gamma-1}{\gamma}}$$

which is the form to be used in all cases where the discharge pressure is to be corrected to constant conditions.

58 In Fig. 16, let *AB* represent the curve of pressure as determined from the original test data, and assume, for instance, that the temperature reading at the intake was higher than the standard, or that the barometer was lower than the standard, test points on the curves being represented by the crosses. The method of determining the pressure for standard conditions then would be to calculate the pressures on the line *CD*, as shown in Fig. 16, the curve being plotted, however, on the original volume lines.

59 When it is impossible to test a blower with the discharge going to the atmosphere, as when a nozzle is used as shown in Fig. 8, the discharge may be measured by obtaining the velocity in the discharge pipe by means of a pitot tube. These measurements are not always reliable, however, as ordinarily made, since great difficulty is experienced in obtaining the true static pressure of a moving fluid, for the reason that the "pressure end" of the tube itself disturbs the stream lines, and hence inaccurate readings are probable.

60 If a pitot tube is used in the ordinary way the pressure end should consist of a thin plate about 4 in. long by about 2 in. wide, having a slit about 1/32 in. wide and 2 in. long at its center. A collecting chamber back of the slit connected to the search pipe tends to equalize the pressure and give a true static pressure reading; but great care is necessary to see that the flat plate is exactly parallel to the stream lines at the point the pressure is desired.

61 A method of determining the flow in pipes which the writer believes is far more accurate and reliable than ordinary pitot tube readings is shown in Fig. 21. A well rounded nozzle having an area of one-fourth to three-fourths that of the pipe is bolted in between two flanges.

62 On the high-pressure side of the nozzle there is a gage *A* and a thermometer (not shown), and a second gage *B* has one leg connected to the high-pressure side of the nozzle and the other to the low-pressure side, thus giving the drop of pressure through the nozzle. A pitot search tube is fitted on the low-pressure side of the nozzle as shown in the figure. This search tube has its opening pointed directly against the flow through the nozzle, and should have a graduated scale so that its position in front of the nozzle can be accurately known.

63 The advantages of this arrangement are that the difficulty of obtaining a true pressure reading is avoided, and a number of checks

may be made on the readings taken, so that should inaccuracies occur for any reason, they can be immediately detected and the trouble located.

64 From Bernouilli's theorem, the sum of the pressure head and velocity head must always be equal, hence, with the search tube E in the position shown in Fig. 21, the pressure differences shown by gages B and C must be equal. Also, if the connection D to gage C be disconnected, the pressure indicated by gage C must equal the pressure shown by gage A. Also, if gage C is connected to the pipe by D, but the pitot tube disconnected, the sum of gages B and C must equal the pressure indicated by gage A. If these various readings check, it follows that the static pressures are being accurately measured and that the pitot tube is pointing directly into the stream.

65 Having thus checked the correctness of the measurements, the jet can be searched by moving the pitot tube across the nozzle at regular intervals and readings taken, from which the velocity at any point can be calculated by the formula

$$V=\sqrt{2g\frac{\gamma}{\gamma-1}RT_1\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right]}$$

ewher

P_1 is the absolute pressure above the nozzle
P_2 is the absolute pressure below the nozzle
T_1 is the observed temperature above the nozzle
V is the velocity in feet per second

With V known and a known area of discharge, the volume passing can be calculated.

66 By this method, if the observations are carefully made, the error should not exceed a very small fraction of 1 per cent. It may not be out of place to add that too great care cannot be taken to make sure that there are no air leaks in the connections to the gages, as even a pinhole may cause a very considerable error in the readings.

67 In the following, in order to show the method of working up a test on a centrifugal blower, the actual readings taken during a 1-hour test on a 30,000 cu. ft. blower have been used. The results have been worked out by seven-place logarithms, using the formulae as given under tests.

68 For practical purposes, however, it is rather inconvenient to work out the values of

$$\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1$$

and

$$\left\{\frac{T_2}{T_2'}\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]+1\right\}^{\frac{\gamma}{\gamma-1}}$$

by logarithms, and therefore the writer has prepared a set of curves for the value of

$$\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1=K$$

for various values of $\frac{P_1}{P_2}$, which may also be used for the solution of the equation for the pressure correction.

APPENDIX

TEST ON WESTINGHOUSE TURBINE BLOWER

Nozzle: 12¼ in.
Static head: $P_1 = 85.6$ in. water $+ 26.66$ in. mercury $= 32.96$ in. mercury.
Barometer $P_2 = 26.66$ in. mercury
Intake temperature: $t_2 = 46.8$ deg. fahr.; $T_2 = 507.5$ deg. fahr. abs.
Discharge temperature: $t_1 = 92.6$ deg. fahr.; $T_1 = 553.3$ deg. fahr. abs.
Total input to motor $= 432.7$ kw.
  input to exciter $= 4.04$ kw.
  input motor only $= 428.66$ kw.
Efficiency motor $= 89.6$ per cent taken from curve including windage
Area of nozzle: 0.817 sq. ft.

$$\left\{\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right\} = 0.06317$$

$$W = \text{work in ft-lb. per lb. of air}$$
$$= 184.754 \times T_2 \times \left\{\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right\} = 5923$$

T'_2 is the absolute temperature at outlet of nozzle
Volume per minute in cubic feet at temperature $T'_2 = \text{Vol. } T'_2$

$$\text{Vol. } T_2' = 109.079 \sqrt{T_1\left[1 - \frac{1}{\left\{\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right\} + 1}\right]} \text{ Area} \times 60 = 30{,}659 \text{ cu. ft.}$$

$$\text{Vol.}_{0}{}^{\circ} = \frac{\text{Vol. } T'_2 \times 520.7}{T_1}\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} = 30{,}670$$

$V_{60\text{F}} = $ Vol. of 1 lb. of air at 60 deg. fahr. and barometer

$$\text{Air h.p.} = \frac{\text{Vol.}_{60}{}^{\circ}}{V_{60\text{F}}} \times \frac{W}{33{,}000} = 373.99$$

To find V_{60}: $pv = RT$ p in lb. sq. ft. $=$ barometer

$$V_{60} = \frac{53.35 \times 520.7}{\text{Bar.}}$$

Barometer $= 26.66$ in. mercury $= 1887$ lb. sq. ft.

Efficiency derived from temperatures:

$$E' = \frac{T_2\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{T_1' - T_2} = 69.997 \text{ per cent}$$

$$T_1 = 553.3$$

$$T_2 = 507.5$$

Efficiency derived from motor input:

Air h.p. = 373.99 Input to motor only = 428.66 kw.

Input to motor and exciter = 432.70 kw.

$$\text{Overall} = \frac{\text{air h.p.}}{\text{total input to motor in h.p.}} = 64.479 \text{ per cent}$$

$$\text{Blower} = \frac{\text{air h.p.}}{(\text{motor input} - \text{exciter input}) \times \text{motor in h.p.}} = 72.641 \text{ per cent}$$

$$P'_1 = 26.7\left[\frac{T_2\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{520.7} + 1\right]^{\frac{\gamma}{\gamma-1}} = 32.837 \text{ in. mercury}$$

$P'_1 - P'_2 = 32.837 - 26.7 = 6.137$ in. mercury

= 48.23 ounces per sq. in. corrected to 26.7 in. barometer and 60 deg. fahr. at the inlet.

RULES FOR CONDUCTING PERFORMANCE TESTS

OF

POWER PLANT APPARATUS

EMBRACING

1 Boilers
2 Reciprocating Steam Engines
3 Steam Turbines
4 Complete Steam Power Plants
5 Pumping Machinery
6 Complete Pumping Plants
7 Compressors, Blowers, and Fans
8 Locomotives
9 Gas Producers
10 Gas and Oil Engines
11 Complete Gas Power Plants
12 Waterwheels

PRELIMINARY REPORT OF POWER TEST COMMITTEE,
CODES OF 1912, THE AMERICAN SOCIETY OF
MECHANICAL ENGINEERS, NEW YORK

TO THE SOCIETY:

In submitting the following preliminary report of the Committee on Power Tests, which was appointed May 4, 1909, with the object of revising and extending the present testing codes of the Society relating to power plant apparatus, the Committee desires a full discussion and criticism of the report by members of the Society and others who are interested in the subject.

The Committee wishes to impress upon those offering suggestions the desirability of stating explicitly in writing the additions, omissions, or changes of language to make the report conform to the views expressed. It will be a great help to the Committee if this request is literally complied with.

It should be borne in mind that the report cannot be expected to deal with all the refinements of laboratory tests or to conform to the methods practised by every individual who engages in testing work; but it is intended to set forth the correct governing principles and serve the practical purposes of the engineering public.

When the preliminary report has been fully discussed and criticised the Committee will make a final revision for publication in the Transactions.

Respectfully submitted,
COMMITTEE ON POWER TESTS
By GEO. H. BARRUS
Vice-Chairman

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FIRST SECTION: GENERAL MATTERS

PART I

INSTRUCTIONS REGARDING TESTS IN GENERAL

INSTRUCTIONS REGARDING TESTS IN GENERAL

1 OBJECT

Ascertain the specific object of the test, and keep this in view not only in the work of preparation, but also during the progress of the test, and do not let it be obscured by devoting too close attention to matters of minor importance. Whatever the object of the test may be, accuracy and reliability must underlie the work from beginning to end.

If questions of fulfillment of contract are involved, there should be a clear understanding between all the parties, preferably in writing, as to the operating conditions which should obtain during the trial, and as to the methods of testing to be followed, unless these are already expressed in the contract itself.

Among the many objects of performance tests, the following may be noted:

Determination of capacity and efficiency, and how these compare with standard or guaranteed results

Comparison of different conditions or methods of operation

Determination of the cause of either inferior or superior results

Comparison of different kinds of fuel

Determination of the effect of changes of design or proportion upon capacity or efficiency, etc.

2 PREPARATIONS

(*A*) *Dimensions*

Measure the dimensions of the principal parts of the apparatus to be tested, so far as they bear on the objects in view, or determine these from correct working drawings. Notice the general features of the same, both exterior and interior, and make sketches, if needed, to show unusual points of design.

The dimensions of the heating surfaces of boilers and superheaters to be found are those of surfaces in contact with the fire or hot gases. The submerged surfaces in boilers at the mean water level should be considered as water-heating surfaces, and other surfaces which are exposed to the gases as superheating surfaces.

In the case of condensers, feedwater heaters, and the like, the outside surfaces are to be taken. In reheaters and steam jackets, the surfaces to be considered are those exposed to the steam of lower pressure.

The dimensions of engine cylinders should be taken when they are cold, and, if extreme accuracy is required, as in scientific investigations, corrections should be applied to conform to the mean working temperature. If the cylinders are much worn, the average diameter should be found. The clearance of the cylinders may be determined approximately from working drawings of the engine. For accurate work, when practicable, the clearance should be determined by the water measurement method. (See Appendix No. 1 for water determination of clearance.)

(*B*) *Examination of Plant*

Make a thorough examination of the physical condition of all parts of the plant or apparatus which concern the object in view, and record the conditions found, together with any points in the matter of operation which bear thereon.

In boilers, for example, examine for leakage of tubes and riveted or other metal joints. Note the condition of brick furnaces, grates and baffles. Examine brick walls and cleaning doors for air leaks, either by shutting the damper and observing the escaping smoke or by candle-flame test. Determine the condition of heating surfaces with reference to exterior deposits of soot and interior deposits of mud or scale.

See that the steam main is so arranged that condensed and entrained water cannot flow back into the boiler.

Ascertain the interior condition of all steam, air, gas, or water cylinders and the condition of their pistons, and of water plungers and impellers, together with the valves and valve-seats belonging thereto. Locate vacuum leaks in exhaust piping, condenser, packings, etc., using vacuum gage or candle-flame test. Examine steam, air, gas, or water piping, traps, drip valves, blow-off cocks, safety valves, relief valves, heaters, etc., and make sure that they do not leak. Determine the condition of the blading, nozzles, and valves in steam turbines, and of buckets, guides and draft-tubes in water turbines.

See Appendices Nos. 2 and 3 for general methods of leakage examination and test.

If the object of the test is to determine the highest efficiency or capacity obtainable, any physical defects, or defects of operation, tending to make the result unfavorable should first be remedied; all fouled parts being cleaned, and the whole put in first-class condition. If, on the other hand, the object is to ascertain the performance under existing conditions, no such preparation is either required or desired.

(*C*) *General Precautions against Leakage*

In steam tests make sure that there is no leakage through blowoffs, drips, etc., or any steam or water connections of the plant or apparatus undergoing test, which would in any way affect the results. All such connections should be blanked off, or satisfactory assurance should be obtained that there is leakage neither out nor in. This is a most important matter, and no assurance should be considered satisfactory unless it is susceptible of absolute demonstration.

See Appendix No. 3 for further details.

(*D*) *Apparatus and Instruments*

Select the apparatus and instruments specified in the Code of Rules applying to the test in hand, locate and install the same, and complete the preparations for the work in view.

The arrangement and location of the testing appliances in every case must be left to the judgment and ingenuity of the engineer in charge, the details being largely dependent upon locality and surroundings. One guiding rule, however, should always be kept in view, viz., see that the apparatus and instruments are substantially reliable, and arrange them in such a way as to obtain correct data.

A summary is given below, embracing the entire list of apparatus and instruments referred to in the various Codes, with descriptions of their leading features, methods of application and use, and, where needed, methods of calibration; these particulars being supplemented in some instances by further descriptions in the Appendix.

(a) *Weighing Scales.* For determining the weight of coal, oil, water, etc., ordinary platform scales serve every purpose. Too much dependence, however, should not be placed upon their reliability without first calibrating them by the use of standard weights, and carefully examining the knife-edges, bearing plates, and ring suspensions, to see that they are all in good order.

Other scales required in connection with test work are small scales for weighing coal-samples used in drying, and laboratory scales for analysis and calorific determinations pertaining to fuels. Such scales should be sensitive to 1/1000 of the quantity weighed.

For testing locomotives and some classes of marine boilers, where room is lacking, sacks or bags are sometimes required to facilitate the handling of coal, the sacks being previously weighed at the time of filling.

(*b*) *Water Weighing and Measuring Apparatus.*

(1) Feedwater. Wherever practicable the feedwater should be weighed, especially for guarantee tests. The most satisfactory and reliable apparatus for this purpose consists of one or more tanks each placed on platform scales, these being elevated a sufficient distance above the floor to empty into a receiving tank placed below,

the latter being connected to the feed pump. Where only one weighing tank is used the receiving tank should be of larger size than the weighing tank, to afford sufficient reserve supply to the pump while the upper tank is filling. If a single weighing tank is used it should preferably be of such capacity as to require emptying not oftener than every five minutes. If two or more are used the intervals between successive emptyings should not be less than three minutes.

In tests of complete steam power plants, where it is required to measure the feedwater without unnecessary change in the working conditions, a water meter may be employed. Meter measurement may also be required in many other cases, such as locomotive and marine service. The reliability of meters should be determined by calibration in place under conditions of use.

If a large quantity of water is to be measured, a rotary, disk, or Venturi meter, or some form of orifice measurement may be employed. In the latter case the orifice should be calibrated under the conditions of use, unless its design is such that standard formulæ and constants may be applied for determining the discharge.

See Appendix No. 4 for methods of calibrating meters and Appendix No. 5 for orifice formula.

(2) Water of Condensation Under Pressure. In measuring jacket water or any supply under pressure which has a temperature exceeding 212 deg. fahr, the water should first be cooled, as may be done by discharging it into a tank of cold water previously weighed, or by passing it through a coil of pipe submerged in running and colder water, preventing thereby the loss of evaporation which occurs when such hot water is discharged into the open air. If such water is untrapped the drain pipe should be provided with a gage glass and the outlet choked so as to keep the water in sight in the glass.

(3) Water Flowing in Streams and Canals. Weirs, current meters, float rods, etc., are required for measuring the water discharged by pumping engines, and that used by waterwheels. (For description see Appendix No. 5.)

(*c*) *Steam Measuring Apparatus.* Various forms of steam meters may be employed for measuring steam, provided such meters are properly calibrated under conditions of use, and the pulsations of pressure, if any, are not serious. For measuring the steam used by the auxiliaries of a steam plant, either individually or collectively, the orifice form of steam meter may be used, consisting of an orifice in a plate inserted between the two halves of a pair of flanges in the pipe through which the steam passes, or placed in a by-pass through which the steam is diverted, with gage pipe on either side for determining the fall of pressure. The quantity of steam represented by the various differences of pressure which occur, may be found by arranging the apparatus so as to draw steam through the orifice, and discharge it into a tank of water resting on platform scales, by which its actual weight in a given time is determined.

A plate ⅛ in. thick containing an orifice 1 in. diameter, with square edges, will discharge the approximate quantities of dry steam per hour given in Table 1, with various pressure drops, the pressure below the orifice being 100 lb. by gage.

TABLE 1 DISCHARGE THROUGH ORIFICE 1 IN. DIAMETER, 100 LB. PRESSURE

Pressure Drop, Lb. Per Sq. In.	Lb. of Dry Steam Per Hour.
½	430
1	615
2	930
3	1200
4	1400
5	1560
10	2180
15	2640
20	3050

The water-glass method affords an approximate means of determining the steam consumption of auxiliaries, and for measuring the leakages of steam and water from the boiler and its connections. (See Appendix No. 3 for description of water glass method.)

(*d*) *Gas and Air Measuring Apparatus.* Apparatus to be used for measuring gas and air embraces gas meters, gasometers, Venturi meters, orifices and pitot tubes. The standard of reference for calibrating apparatus for this purpose is the gasometer. If a pitot tube is used, and this instrument cannot be calibrated by actual measurement, the constants employed should be those obtained from a similar instrument which has been calibrated by actual reference to gasometer measurement under as nearly as possible the same conditions.

Directions for using the pitot tube for gas and air measurement are given in Appendix No 6.

In this connection attention is directed to the electric method of gas measurement proposed by Mr. C. C. Thomas and described in his paper in Volume No. 31 of the Transactions,[1] also to orifice measurements described by R. J. Durley, in Volume 27.[2]

(*e*) *Anemometers.* For determining the approximate quantity of air discharged under light pressures by fans or blowers, or the amount of air used in the combustion of fuels, an anemometer of the fanwheel type may be used, provided all the air can be passed through one or more openings of suitable size in which the anemometer may be placed. These openings should preferably be of such size (say at least 12 in. sq.) that the resistance interposed by the instrument (the fan being about 2½ in. in diameter) may be neglected. The instrument, when in use, can best be supported on the end of a small rod and moved slowly across the open end, also up and down, so as to obtain an average for the whole area. In the case of very large openings the area should be subdivided into a number of equal parts, the velocity determined for each subdivision, and the results averaged.

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[1] C. C. Thomas, An Electric Gas Meter. Trans. Am. Soc. M. E., vol. 31, p. 655.

[2] R. J. Durley, On the Measurement of Air Flowing into the Atmosphere through Circular Orifices in Thin Plates and under Small Differences of Pressure. Trans. Am. Soc. M. E., vol. 27, p. 193.

> The anemometer is usually calibrated by mounting it on the end of a long and light rod which can be revolved around a central point and its reading compared with the velocity of the rod when moved in still air. Where suitable means can be provided for calibrating the anemometer under approximately its working conditions, such calibration is much to be preferred.

The quantity of air supplied for combustion in boiler and other furnaces may be determined from the analysis of the products of combustion using the formula given in the Boiler Code, p. 1724.

(f) *Screens for Sizing Coal.* The dimensions of screen openings to be used for sizing anthracite coals are given in Table 2; the sizes in each case being the opening through which the specified grade will pass and that over which it will be carried without passing through. The openings referred to are circular.

TABLE 2 ANTHRACITE COAL SIZES

Name	Diameter of opening through or over which coal will pass, in.		Name	Diameter of opening through or over which coal will pass, in.	
	Through	Over		Through	Over
Broken	4 1/2	3 1/4	No. 1 Buckwheat	9/16	5/16
Egg	3 1/4	2 5/16	No. 2 Buckwheat	5/16	3/16
Stove	2 5/16	1 5/8	No. 3 Buckwheat	3/16	3/32
Chestnut	1 5/8	7/8	Culm	3/32	...
Pea	7/8	9/16			

The sizes and grades of bituminous and semi-bituminous coals vary so much according to kind and locality that there are no standards of size for these coals which are generally recognized. (For bituminous coal sizes see Appendix No. 7.)

(g) *Pressure Gages.* For determining steam pressure the steam gages belonging to the plant may be used, provided they are compared with a standardized gage of the spring or mercury type and verified, due allowance being made for the head of water standing in the connecting pipe.

For determining pressure below the atmosphere, a U-tube mercury gage may be employed or a spring gage may be used, provided it is referred to a standard and corrected for water in the connecting pipe. In cases where extremely high vacuums are to be measured, as in turbine practice, the absolute-pressure gage is useful, provided the exhaustion is complete and afterwards no air is admitted.

For determining steam pressure on the two sides of an orifice, two gages should be used which are carefully graduated to single pounds, or, what is better, one gage should be used and this piped up so as to connect at will to either side of the orifice. A differential gage may also be employed, indicating at once the pressure drop.

For determining the water pressure in the force main of a pumping engine, the gage should be one which is sensitive to changes amounting to ½ per cent. of the pressure indicated. If such a gage is not a part of the equipment of the plant, a special test gage should be attached.

> Between the gage and the force main a small reservoir having an air chamber should be interposed, in the manner shown in Appendix No. 8, so as to prevent undue fluctuations of the gage and allow the gage cock to be run wide open. By means of a gage glass on the side of the chamber and an air valve, the average water level may be adjusted to the height of the center of the gage, and correction for this element of variation avoided. If not thus adjusted, the reading is to be referred to the level shown, whatever this may be.

For calibrating gages indicating pressures above the atmosphere, the dead-weight testing apparatus which is manufactured by many of the prominent gage makers may be employed as a standard of comparison. It consists of a vertical plunger nicely fitted to a cylinder containing oil or glycerine, through the medium of which the pressure is transmitted to the gage. The plunger is surmounted by a circular stand on which weights may be placed, and by means of which any desired pressure can be secured. The total weight, in pounds, on the plunger at any time, divided by the average area of the plunger and of the bushing which receives it, in square inches, gives the pressure in pounds per square inch.

Another standard of comparison is the mercury column. If this instrument is used, assurance must be had that it is properly graduated with reference to the ever-varying zero point; that the mercury is pure, and that the proper correction is made for any difference of temperature that exists, compared with the temperature at which the instrument was graduated.

For pressures below the atmosphere, an air pump or some other means of producing a vacuum is required, and reference must be made to a mercury gage. Such a gage may be a U-tube having a length of 30 in. or so, with both arms properly filled with pure mercury.

> The practice of choking the gage cock to reduce fluctuations of the gage pointer, if carried too far, is objectionable, especially when there is leakage around the plug. The pressure indicated under these circumstances is less than the true pressure. Before reading a gage of

the spring type, the free working of the mechanism within should be assured by tapping the outside of the gage and thereby slightly moving the pointer.

(*h*) *Thermometers.* Thermometers should be of the kind having graduations marked on the glass stem. Those used for temperatures above the boiling point of mercury (say 450 deg.—500 deg.) should have nitrogen in the top of the bore. They should also have a small safety bulb at the top. Thermometers constructed in this way can be used satisfactorily up to 1000 deg. fahr.

Thermometers which are used for important data should be calibrated before and after a test, by reference to standard thermometers.

Standard thermometers are those which indicate 212 deg. fahr. in steam escaping from boiling water at the normal barometric pressure of 29.92 in. (referred to 32 deg.), the whole stem up to the 212 deg. point being surrounded by the steam; and which indicate 32 deg. fahr. in melting ice, the stem being likewise completely immersed to the 32 deg. point; and which are calibrated for points between and beyond these two reference marks For temperatures between 212 deg. and 400 deg. fahr., the comparison of the thermometer should be made with the temperature given in Marks and Davis Steam Tables, the method required being to place it in a thermometer well surrounded by saturated steam under sufficient pressure to give the desired temperature. The pressure should be determined by a correct gage, and the thermometer should be immersed to the same extent as it is under its working condition.

A thermometer well consists of a hollow plug threaded at the upper end and screwed into a threaded hole in the top of a horizontal pipe, the lower part extending vertically into the interior of the pipe as far, if practicable, as the center. The inside diameter should be slightly larger than the outside diameter of the thermometer tube and the well should be filled with mercury or high-grade mineral oil for temperatures below 500 deg., and with soft solder for higher temperatures.

For superheated steam the immersed portion should be fluted so as to increase the area of the absorbing surface.

Thermometers are so readily broken that it is desirable in important tests to have a sufficient number on hand that in case of accident the readings will not be interrupted. These spare thermometers should preferably be calibrated.

(*i*) *Barometers.* For important or extremely accurate steam tests and for gas engine tests the pressure of the atmosphere should be taken, either by a mercurial or Aneroid barometer, and the reading from this instrument, reduced to pounds pressure per square inch, should be employed in determining the absolute steam pressure. In many cases it is sufficient to refer to the daily records of the nearest station of the Government Weather Bureau. Aneroid barometers may be readily calibrated by comparing them with a mercury barometer, making proper temperature corrections.

(*j*) *Hygrometers.* In tests where the hygrometric conditions of the atmosphere play an important part, it is necessary to use a hygrometer. The ordinary instrument consisting of a wet and dry bulb thermometer is suitable for the purpose. The hygrometric tables of the Government which may be obtained from the Weather Bureau, U. S. Dept. of Agriculture, Washington, D. C., can be referred to in making the required computations. See Appendix No. 9.

(*k*) *Pyrometers.* Metallic pyrometers used for determining high temperatures must be handled cautiously owing to the difficulty of exposing the whole of the stem to the current of gas, the temperature of which is to be determined. Electric pyrometers either of the thermo-couple or resistance type are satisfactory for this work where the temperatures to be observed are not too high, say not over 2000 fahr. Instruments of this kind can readily be calibrated by comparing them at low ranges of temperature under proper conditions with a mercurial thermometer, (both being placed for example in a current of hot air the temperature of which is under control), and their accuracy at higher temperatures judged by plotting the results on a chart, finding the curve of error, and continuing the curve to the higher ranges desired. For extremely high temperatures such as that of a boiler or producer furnace, the optical, pneumatic, and radiation pyrometers may be used. See Appendix No. 10.

(*l*) *Draft Gages.* The simplest form of draft gage is the ordinary U-tube. When the tube is kept clean and the two legs are close together with the scale extending at least to the center of each leg, it gives satisfactory indications. For measuring small amounts of draft some form of multiplying gage may be employed. One of the simplest multiplying instruments consists of a U-tube in which one leg is inclined from the horizontal and the amount of multiplication varies inversely as the angle of inclination, the tube being filled with a light mineral oil. Various satisfactory instruments having the multiplying feature are on the market. These can readily be calibrated by comparison with the simple U-tube gage when indicating a high draft, say one inch or more. It is preferable to use alcohol instead of water in the U-tube, and make allowance for the difference of specific gravity.

(*m*) *Steam Calorimeters.* The most satisfactory instruments for determining the amount of moisture in steam are calorimeters that operate upon the throttling principle, or that combine the throttling and separating principles; an orifice being used of such size as to throttle to atmospheric pressure, with two thermometers, one showing the temperature above the orifice and the other that below it. If no commercial make of calorimeter is available on a test, an instrument of the throttling type can be made of pipe fittings as shown in Appendix No. 11.

Directions for applying sampling nozzles are given in Part III, Page 1713.

Further references to Steam Calorimeters are included in Appendix No. 11.

(*n*) *Coal Calorimeters.* To determine the total heat of combustion of a sample of coal or other fuel, the best form of calorimeter to use is one in which the fuel is burned in an atmosphere of oxygen gas. The Mahler type of calorimeter is recognized as the most complete and accurate apparatus of this kind. Where the engineer conducting a test does not have this instrument, or some other reliable calorimeter at hand, the heat units can be determined by sending samples to a testing laboratory where such instruments are used.

For description of one form of the Mahler Calorimeter see Appendix No. 12.

(*o*) *Gas Calorimeters.* The total heat of combustion of gas should be found by burning the gas in the Junker calorimeter, described in Appendix No. 13.

(*p*) *Coal Analysis Apparatus.* The analyses commonly made are what are termed "proximate" analyses. For complete determinations of the quality of coal, it is necessary also to make the "ultimate" analysis. Approved methods of analysis are briefly described in Appendix No. 14.

(*q*) *Gas Analysis Apparatus.* For determining the composition of flue gases in ordinary boiler work one of the simplest and most convenient instruments is the Orsat apparatus. This instrument can readily be used by the person conducting a test, or by some assistant whom he directs.

For determining the hydrogen and other unburned combustible matter in the flue gases, and for general gas analysis, the Hempel apparatus, or some modification thereof, is required and the work should be entrusted to a person who is familiar with all phases of the subject. These instruments are briefly described in Appendix No. 15.

(*r*) *Appliances and Methods Pertaining to Smoke Determination.* No wholly satisfactory methods for either quantitative or qualitative smoke determinations have yet come into use, nor have any reliable methods been established for definitely fixing even the relative density of the smoke issuing from chimneys at different times. One method commonly employed which answers the purpose fairly well, is that of making frequent visual observations of the chimney at intervals of one minute or less for a period of one hour and recording the observed characteristics according to the degree of blackness and density, and giving to the various degrees of smoke an arbitrary percentage value rated in some such manner as that expressed in Table 3.

TABLE 3 SMOKE PERCENTAGES

Dense black	100
Medium black	80
Dense gray	60
Medium gray	40
Light gray	20
Very light	5
Trace	1
Clear chimney	0

The color and density of smoke depend somewhat on the character of the sky or other background, and on the air and weather conditions obtaining when the observation is made, and these should be given due consideration in making comparisons. Observations of this kind are also subject to personal errors and errors of judgment. Nevertheless, these methods are useful, especially when the results are plotted, according to the percentage scale determined on, so that a graphic representation of the changes can be shown.

Various forms of charts and clouded glass arrangements for comparing and fixing smoke densities have been proposed, and to some extent used, but these have proved more or less unsatisfactory and they are subject to personal errors, and to sky, wind, and weather conditions, the same as the simpler method above described.

Among the chart methods referred to, the use of the Ringelmann smoke chart is perhaps the most familiar. This is shown in Appendix No. 16.

Another method of smoke determination consists in the use of a narrow flat metal plate suspended in the flue, the character of the smoke being indicated by the amount and quality of the soot and dust deposited upon the plate in a given time. This method, like others, is useful in furnishing a means of comparison in different cases rather than a means of exact determination.

See Appendix No. 17 for further description.

(*s*) *Indicators.* To determine the amount of power developed by an engine, whether a steam engine or an internal-combustion engine, the instrument required is the widely-used steam engine indicator. One or more of these instruments is attached to the cylinder or cylinders of the engine, and operated from the cross-head or main shaft by the use of proper driving rig. As to the selection of the make of instrument, it should be one which is in all respects of first-class construction.

For fuller particulars see Appendix No. 18.

(*t*) *Planimeters.* To determine the area of indicator diagrams from which to ascertain the mean effective pressure, it is convenient to use some form of planimeter. The simplest and probably the most desirable instrument is the familiar Polar planimeter, in which the area is registered in square inches.

It is desirable to calibrate a planimeter from time to time by running it over a figure having a known area, such as a right angle triangle of say 4 in. in length and 2 in. in height, observing whether it checks with the computed area.

(*u*) *Tachometers and Other Speed Measuring Apparatus.* For determining the speed of revolution of an engine shaft, especially where the speed exceeds 300 r.p.m., a convenient instrument is a tachometer which continuously indicates on a dial the number of turns per minute. This instrument can be arranged to have a permanent location and to be operated continuously when the engine is running, or it can be a portable instrument which is held in the hand and applied for the time being to the end of the shaft. These instruments are of two general classes, viz: fly-ball and electro-magnetic.

These instruments should be calibrated by comparison with the record obtained by counting with the watch and a speed recorder or indicator, the number of turns per minute.

The determination of variation of speed during a single revolution, or the effect due to sudden changes of the load, is desirable, especially in engines driving electric generators used for lighting purposes. There is no recognized standard method of making such determinations, and if they are desired the method may be devised to suit the requirements.

One method suggested for determining the instantaneous variation of speed which accompanies a change of load, is described as follows: A screen containing a narrow slot is placed on the end of a bar and vibrated by means of an electric current. A corresponding slot in a stationary screen is placed parallel and nearly touching the vibrating screen, and the two screens are placed a short distance from the flywheel of the engine in such a position that the observer can look through the two slots in the direction of the spokes of the wheel. The vibrations are adjusted so as to conform to the frequency with which the spokes of the wheel pass the slots. When this is done the observer viewing the wheel through the slots sees what appears to be a stationary flywheel. When a change in the velocity of the flywheel occurs, the wheel appears to revolve either backward or forward according to the direction of the change. By careful observations of the amount of this motion, the angular change of velocity during any given time is revealed.

(*v*) *Friction Brakes.* The power delivered by an engine may be determined by the application of a Prony brake to the rim of the flywheel. The friction device may consist of a simple band or rope, a number of ropes, or a series of blocks, encircling the wheel. Weighing scales either of the platform or spring type are provided for showing the strain. For long runs the wheel is made with interior flanges for holding water to keep the rim cool.

The most satisfactory brake for absorbing and measuring power is some form of water friction brake. The advantage of a water brake is that it can be employed equally well for large or small amounts of power, and it is necessarily kept cool by the water upon which it depends for its operation.

Several brakes are described in Appendix No. 19.

(*w*) *Dynamometers.* Dynamometers furnish means for determining the amount of power delivered by an engine under working conditions. In the case of a mill engine it is the power transmitted from the main shaft of the engine to the driving shaft of the mill. If this power is carried through a belt, the dynamometer measures the net amount of force transmitted. In a marine engine driving a screw propeller through a long shaft, the dynamometer shows the torsional strain on the shaft at a point as near as practicable to the engine. In a locomotive the dynamometer measures the amount of pull on the draw-bar through which the power is transmitted to the first car of the train.

See Appendix No. 20 for details regarding dynamometers.

(x) *Electrical Instruments.* The output of an engine driving an electric generator should be determined by the use of a set of electrical instruments supplementary to the switchboard equipment, and connected to the generator, so as to be thrown in and out of circuit by means of switches. These instruments should include the shunts, transformers, or multipliers, which belong to them, and should be such as have been verified by comparison with known standards based on those of the U. S. Bureau of Standards at Washington. The instruments selected should be as nearly "dead-beat" as practicable, that is, the pointer should come to rest at once, after changes of load.

The instruments required for a direct-current generator, where the load is substantially constant, embrace a single ammeter and a single voltmeter. Where the load is rapidly fluctuating, a watt-hour meter is also needed.

Those required for a single-phase alternating-current generator carrying a substantially constant load, embrace one ammeter, one voltmeter, and one single-phase wattmeter. For a rapidly fluctuating load a watt-hour meter should also be used. For a two-, or three-phase, alternating-current generator, with substantially constant load, one ammeter and one voltmeter, each arranged for ready connection at will to any phase, and two single-phase wattmeters, are required. Where the load is rapidly fluctuating a polyphase watt-hour meter is also needed.

In the case of an alternating-current generator an additional voltmeter and ammeter should be used for measuring the current for excitation, unless the main instruments are connected so as to register the net output.

The instruments should be so located with reference to the generator, switchboard, and neighboring conductors, as to avoid the disturbing effects due to stray fields. If the switchboard equipment is complicated, the wiring should be connected under the supervision of one who is familiar with all the details of the switchboard, its connections, and the surrounding cables.

Watt-hour meters should be verified in place and under working conditions. Directions for calibrating these meters and further details regarding the application and use of electrical instruments, determination of power factor, etc., are given in Appendix No. 21.

(y) *Water Rheostats.* For tests of engines driving electric generators under certain fixed loads it is often necessary to regulate the output of current by the use of a water rheostat. One of the simplest and most effective forms of this apparatus for direct-current work consists of a coil of wire wound spirally on a wooden reel immersed in a stream of running water, or in a tank of some natural body of water where circulation can be provided for, arranged with a system of switches so that more or less of this resistance can be turned on as may be required. A No. 12 B. G. iron wire 2000 ft. long, thus used, will handle a current of 1600 amperes at 250 volts.

Other rheostats are described in Appendix No. 22.

(z) *Steam Tables.* Quantities depending upon the properties of saturated steam which are used throughout the Codes, such as B.t.u. per pound of steam, temperatures corresponding to various pressures, etc., are based on Marks and Davis's tables; and those involving the specific heat of superheated steam, on Knoblauch and Jakob's experiments.

3 MISCELLANEOUS INSTRUCTIONS

The person in charge of a test should have the aid of a sufficient number of assistants, so that he may be free to give special attention to any part of the work whenever and wherever it may be required. He should make sure that the instruments and testing apparatus continually give reliable indications, and that the readings are correctly recorded. He should also keep in view, at all points, the operation of the plant or part of the plant under test and see that the operating conditions determined on are maintained and that nothing occurs, either by accident or design, to vitiate the data. This last precaution is especially needed in guarantee tests.

Before a test is undertaken, it is important that the boiler, engine, or other apparatus concerned, shall have been in operation a sufficient length of time to attain working temperatures and proper operating conditions throughout, so that the results of the test may express the true working performance.

It would, for example, be manifestly improper to start a test for determining the maximum efficiency of an externally fired boiler with brick setting, until the boiler had been at work a sufficient number of days to dry out thoroughly and heat the brick work to its working temperature; and likewise improper to begin an engine test for determining the performance under certain prearranged conditions until those conditions had become established by a suitable preliminary run.

An exception should be noted where the object of the test is to obtain the working performance, including the effect of preliminary heating, in which case all the conditions should conform to those of regular service.

In preparation for a test to demonstrate maximum efficiency, it is desirable to run preliminary tests for the purpose of determining the most advantageous conditions.

4 OPERATING CONDITIONS

In all tests in which the object is to determine the performance under conditions of maximum efficiency, or where it is desired to ascertain the effect of predetermined conditions of operation, all such conditions which have an appreciable effect upon the efficiency should be maintained as nearly uniform during the trial as the limitations of practical work will permit. In a stationary steam plant, for example, where maximum efficiency is the object in view, there should be uniformity in such matters as steam pressure, times of firing, quantity of coal supplied at each firing, thickness of fire, and in other firing operations; also in the rate of supplying the feed-water, in the load on the engine or turbine, and in the operating conditions throughout. On the other hand, if the object of the test is to determine the performance under working conditions, no attempt at uniformity is either desired or required unless this uniformity corresponds to the regular practice, and when this is the object the usual working conditions should prevail throughout the trial.

5 RECORDS

A log of the data should be entered in notebooks or on blank sheets suitably prepared in advance. This should be done in such manner that the test may be divided into hourly periods, or if necessary, periods of less duration, and the leading data obtained for any one or more periods as desired, thereby showing the degree of uniformity obtained.

The readings of the various instruments and apparatus concerned in the test other than those showing quantities of consumption (such as fuel, water, and gas), should be taken at intervals not exceeding half an hour and entered in the log. Whenever the indications fluctuate, the intervals should be reduced according to the extent of the fluctuation. In the case of smoke observations, for example, it is often necessary to take observations every minute, or still oftener, continuing these throughout the period covering the range of variations.

Make a memorandum of every event connected with the progress of a test, however unnecessary at the time it may appear. A record should be made of the exact time of every such occurrence and the time of taking every weight and every observation. For the purpose of identification the signature of the observer and the date should be affixed to each log sheet or record.

In the simple matter of weighing coal by the barrow-load, or weighing water by the tank-full, which is required in many tests, a series of marks, or tallies, should never be trusted. The time each load is weighed or emptied should be recorded. The weighing of coal should not be delegated to unreliable assistants, and whenever practicable, one or more men should be assigned solely to this work. The same may be said with regard to the weighing of feedwater.

6 DATA REPRESENTED GRAPHICALLY

If it is desired to show the uniformity of the data at a glance the whole log of the trial should be plotted on a chart, using horizontal distances to represent times of observation, and vertical distances on suitable scales to represent various data as recorded. Such a chart showing log of a boiler test is illustrated in Appendix No. 23.

It is instructive to plot the leading data on such a chart while the test is in progress.

7 REPORT

The report of a test should present all the leading facts bearing on the design, dimensions, condition, and operation of the apparatus tested, and should include a description of any other apparatus and auxiliaries concerned, together with such sketches as may be needed for a clear understanding of all points under consideration. It should state clearly the object and character of the test, the methods followed, the conditions maintained, and the conclusions reached, closing with a tabular summary of the principal data and results.

PART II

STANDARDS RELATING TO CAPACITY, EFFICIENCY, AND ECONOMY, AND RULES FOR RESERVE CAPACITY

STANDARDS

The standard units on which to base the various measures of capacity, and the standard forms of expressing efficiency and economy to which the codes apply, are assembled in Tables 4 and 5.

TABLE 4 STANDARD UNITS OF CAPACITY

a Boilers*	One pound of water evaporated into dry steam from and at 212 deg. per hour.	
b Reciprocating Steam Engines	One indicated horsepower developed in the main cylinders	i.h.p.
	One brake horsepower delivered by the main shaft	br.h.p.
c Steam Turbines	One brake horsepower delivered by the main shaft	br.h.p.
d Turbo-generators, (including engine-driven generators)	One kilowatt-hour delivered at the bus-bar,† not including exciter output‡	kw-hr.
e Pumping Machinery	One gallon of water discharged to the force main in 24 hr.	
	One gallon of water discharged per min.§	gal.pr.min.
	One water horsepower delivered to the force main, based on the total head including suction	w.h.p.
f Air Machinery	One cu. ft. of air at 62 deg. and 30 in.¶	cu.ft.
	One air horsepower	air h.p.
g Locomotives	One indicated horsepower developed in the main cylinders	i.h.p.
	One dynamometer horsepower delivered to the draw-bar	dyn. h.p.

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* A subsidiary unit which may be used for stationary boilers is a "Boiler Horsepower," or 34½ lb. of water evaporated from and at 212 deg. per hour, i. e., from water at 212 deg. into steam at the same temperature. Electrical Engineers have suggested a unit termed "Myrowatt," which differs but little from boiler horsepower when expressed in B.t.u. per hour.

† It is assumed that the drop in voltage between generator terminals and switchboard is not over one-half of one per cent.

‡ If the exciter current is taken from an outside source the kw. thus supplied are to be deducted from the total output.

§ This unit applies to small pumps and some classes of large sized pumps.

¶ 30-in. barometer is referred to a temperature of 62 deg.; or 29.92-in. referred to 32 deg.

h Gas Producers.....	One pound of dry fuel consumed per hour.	
i Gas and Oil Engines.	One brake horsepower delivered by the main shaft..........................	br.h.p.
j Waterwheels	One brake horsepower delivered by the main shaft..........................	br.h.p.

TABLE 5 STANDARDS OF EFFICIENCY AND ECONOMY

a Boilers	Relation between B.t.u. absorbed by boiler per lb. of coal fired and calorific value of 1 lb. coal. (Efficiency of boiler furnace and grate.) Relation between B.t.u. absorbed by boiler, per lb. of combustible burned and calorific value of 1 lb. combustible. (Efficiency of boiler and furnace.)
b Reciprocating Steam Engines	(1) B.t.u. per i.h.p.-hr. (2) B.t.u. per brake h.p.-hr. (3) Thermal efficiency ratio referred to i.h.p. (4) Thermal efficiency ratio referred to br. h.p. (5) Lb. of dry steam per i.h.p.-hr. (6) Lb. of dry steam per br. h.p.-hr.
c Steam Turbines....	(1) B.t.u. per br. h.p.-hr. (2) Thermal efficiency ratio. (3) Lb. of dry steam per br. h.p.-hr.
d Turbo-generators, (including engine-driven generators)	(1) B.t.u. per kw-hr. (2) Thermal efficiency ratio. (3) Lb. of dry steam per kw-hr.
e Pumping Engines...	(1) Ft-lb. of work per million B.t.u. (2) Ft-lb. of work per 1000 lb. dry steam. (3) Ft-lb. of work per 100 lb. dry fuel.
f Air Machinery......	(1) B.t.u. per net air h.p. per hr. (2) Lb. of dry steam per net air h.p. per hr. (3) Lb. of dry steam per 1000 cu. ft. of free air compressed to 100 lb. gage pressure reduced to atmospheric temperature.
g Locomotives	(1) Lb. of dry fuel per i.h.p.-hr. (2) Lb of dry fuel per dyn. h.p.-hr. (3) Lb. of dry steam per i.h.p.-hr. (4) Lb. of dry steam per dyn. h.p.-hr. (5) Lb. of dry fuel per ton-mile.
h Gas Producers.....	Relation between B.t.u. of the gas output per lb. of fuel charged and calorific value of 1 lb. of fuel.
i Gas and Oil Engines.	(1) B.t.u. per brake h.p.-hr. (2) Thermal efficiency ratio. (3) Lb. of dry fuel per br. h.p.-hr. (4) Cu. ft. of gas per br. h.p.-hr.
j Waterwheels	Relation between brake h.p. and potential h.p. of total water used.

k Steam Power Plants.	(1) Lb. of dry fuel per i.h.p.-hr., main engine.
	(2) Lb. of dry fuel per i.h.p.-hr., auxiliaries.
	(3) Lb. of dry fuel per i.h.p.-hr., whole plant.
	(4) Lb. of dry steam per i.h.p.-hr., main engine.
	(5) Lb. of dry steam per i.h.p.-hr., auxiliaries.
	(6) Lb. of dry steam per i.h.p.-hr., whole plant
l Electric Power Plants	(1) Lb. of dry fuel per kw-hr., main engine or turbine
	(2) Lb. of dry fuel per kw-hr., auxiliaries.
	(3) Lb. of dry fuel per kw-hr., whole plant.
	(4) Lb. of dry steam per kw-hr., main engine or turbine.
	(5) Lb. of dry steam per kw-hr., auxiliaries.
	(6) Lb. of dry steam per kw-hr., whole plant.
m Pumping Engine Plants	(1) Ft-lb. of work per million B.t.u.
	(2) Ft-lb. of work per 100 lb. dry fuel.
	(3) Ft-lb. of work per 1000 lb. dry steam.
n Gas Power Plants..	(1) Lb. of dry fuel per brake h.p.-hr.
	(2) Cu. ft. of gas per br. h.p.-hr.

The i.h.p. and brake h.p. in this table refer to that of the main engine, turbine, or waterwheel, and the kw. to the current measured at the bus-bar, not including exciter current.

Contracts for power plant apparatus should specify the leading dimensions of the apparatus and its rated capacity, expressed in the units given in Table 4. If a specific guarantee of capacity is made, either working capacity or maximum capacity, the operating conditions under which the guarantee is to be met should be clearly set forth; such, for example, as steam pressure, speed, vacuum, quality of fuel, force of draft, etc. Likewise if a contract contains a guarantee of economy all the conditions should be fully specified.

The commercial rating of capacity determined on for power plant apparatus, whether for the purpose of contracts for sale, or otherwise, should be such that a sufficient reserve capacity beyond the rating is available to meet the contingencies of practical operation; such contingencies, for example, as the loss of steam pressure and capacity due to cleaning fires, inferior coal, oversight of the attendants, sudden demand for an unusual output of steam or power, etc. To secure this end, the following requirements should be met:

(*a*) *Boilers.* The reserve capacity of a boiler should be at least one-third of the commercial rating, when using coal which is regarded as a standard where the boiler is located, the fire being crowded, and the draft between the damper and the boiler being at least ½-in. water column.

A sufficient amount of grate surface should be provided in such a boiler to develop the rated capacity with the coal and draft named without crowding the fire.

(*b*) *Reciprocating Steam Engines and Steam Turbines.* The reserve capacity of a steam engine or steam turbine at a stipulated steam pressure should be such as to allow a drop of at least 15 per cent in the pressure without material reduction in the normal speed. It should also allow an overload at the specified pressure amounting to at least 25 per cent of the rated power.

(*c*) *Pumping Engines.* The reserve capacity of a pumping engine should be such as to permit a drop in the steam pressure of at least 15 per cent without sensible reduction in the quantity of water discharged at its rated capacity, and to allow an increase in power sufficient to discharge 20 per cent more water than the rated amount.

(*d*) *Gas Producers.* The reserve capacity of a gas producer should be such that when forced it will burn in a given time 20 per cent more coal of the quality agreed upon than the rated capacity.

(*e*) *Gas and Oil Engines.* The reserve capacity of an internal-combustion engine should be such that when supplied with gas of the kind and quality which it is designed to use, it should develop at least 20 per cent more power than the commercial rating.

(*f*) *Waterwheels.* The reserve capacity of a waterwheel should be at least 10 per cent more than the commercial rating at the specified head, the buckets in the wheel being clean and the flow of water unobstructed.

PART III

RULES FOR SAMPLING AND DRYING COAL AND ASH AND SAMPLING STEAM

RULES FOR SAMPLING AND DRYING COAL AND ASH AND SAMPLING STEAM

(*A*) *Sampling and Drying Coal*

Select a representative shovelful from each barrow-load as it is drawn from the coal pile or other source of supply, and store the samples in a cool place in a covered metal receptacle. When all the coal has thus been sampled, break up the lumps, thoroughly mix the whole quantity, and finally reduce it by the process of repeated quartering and crushing to a sample weighing about 5 lb., the largest pieces being about the size of a pea. From this sample two 1-qt. air-tight glass fruit jars, or other air-tight vessels, are to be promptly filled and preserved for subsequent determinations of moisture, calorific value, and chemical composition. These operations should be conducted where the air is cool and free from drafts

When the sample lot of coal has been reduced by quartering to say 100 lb., a portion weighing say 15 to 20 lb. should be withdrawn for the purpose of immediate moisture determination. This is placed in a shallow iron pan and dried on the hot iron boiler flue for at least 12 hours, being weighed before and after drying on scales reading to quarter ounces.

The moisture thus determined is approximately reliable for anthracite and semi-bituminous coals, but not for coals containing much inherent moisture. For such coals, and for all absolutely reliable determinations the method to be pursued is as follows:

> Take one of the samples contained in the glass jars, and subject it to a thorough air drying, by spreading it in a thin layer and exposing it for several hours to the atmosphere of a warm room, weighing it before and after, thereby determining the quantity of surface moisture it contains. Then crush the whole of it by running it through an ordinary coffee mill or other suitable crusher adjusted so as to produce somewhat coarse grains (less than 1-16 in.), thoroughly mix the crushed sample, select from it a portion of from 10 to 50 grams[1], weigh it in a balance which will easily show a variation as small as 1 part in 1000, and dry it for one hour in an air or

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[1] Say ½ oz. to 2 oz.

sand bath at a temperature between 240 and 280 deg. fahr. Weigh it and record the loss, then heat and weigh again until the minimum weight has been reached. The difference between the original and the minimum weight is the moisture in the air-dried coal. The sum of the moisture thus found and that of the surface moisture is the total moisture.

(*B*) *Sampling Ashes and Refuse*

The method above described may also be followed for obtaining a sample of the ashes and refuse.

(*C*) *Sampling Steam*

Construct a sampling pipe or nozzle made of ½-in. iron pipe and insert it in the steam main at a point where the entrained moisture is likely to be most thoroughly mixed. The inner end of the pipe, which should extend nearly across to the opposite side of the main, should be closed and the interior portion perforated with not less than twenty ⅛-in. holes equally distributed from end to end and preferably drilled in irregular or spiral rows, with the first hole not less than half an inch from the wall of the pipe.

The sampling pipe should not be placed near a point where water may pocket or where such water may affect the amount of moisture contained in the sample. Where non-return valves are used, or where there are horizontal connections leading from the boiler to a vertical outlet, water may collect at the lower end of the uptake pipe and be blown upward in a spray which will not be carried away by the steam owing to a lack of velocity. A sample taken from the lower part of this pipe will show a greater amount of moisture than a true sample. With goose-neck connections a small amount of water may collect on the bottom of the pipe near the upper end where the inclination is such that the tendency to flow backward is ordinarily counterbalanced by the flow of steam forward over its surface; but when the velocity momentarily decreases the water flows back to the lower end of the goose-neck and increases the moisture at that point, making it an undesirable location for sampling. In any case it should be borne in mind that with low velocities the tendency is for drops of entrained water to settle to the bottom of the pipe, and to be temporarily broken up into spray whenever an abrupt bend or other disturbance is met.

If it is necessary to attach the sampling nozzle at a point near the end of a long horizontal run, a drip pipe should be provided a short distance in front of the nozzle, preferably at a pocket formed by some fitting and the water running along the bottom of the main drawn off, weighed, and added to the moisture shown by the calorimeter; or, better, a steam separator should be installed at the point noted.

(D) *General Location of Sampling Pipe and Thermometer-Well*

In testing a stationary boiler the sampling pipe should be located as near as practicable to the boiler, and the same is true as regards the thermometer well when the steam is superheated. In an engine or turbine test these locations should be as near as practicable to the throttle valve. In the test of a plant where it is desired to get complete information, especially where the steam main is unusually long, sampling nozzles or thermometer wells should be provided at both points, so as to obtain data at either point as may be required.

In a locomotive, the calorimeter should be attached either to the steam dome where it may be connected to the throttle opening, or to the steam passage in the saddle casting on one side.

SECOND SECTION: INDIVIDUAL CODES

PART IV

BOILER CODE

RULES FOR CONDUCTING EVAPORATIVE TESTS OF BOILERS

1 OBJECT AND PREPARATIONS

Determine the object, take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as pointed out in the general instructions given on pages 1689 to 1714, and make preparations for the test accordingly.

2 FUEL

Determine the character of fuel to be used.[1] For tests of maximum efficiency or capacity of the boiler to compare with other boilers, the coal should be of some kind which is commercially regarded as a standard for the locality where the test is made.

In the Eastern States the standards thus regarded for semi-bituminous coals are Pocahontas (Va. and W. Va.) and New River (W. Va.); for anthracite coals those of the No. 1 buckwheat size, fresh-mined, containing not over 13 per cent ash by analysis; and for bituminous coals, Youghiogheny and Pittsburg coals. In some sections east of the Allegheny Mountains the semi-bituminous Clearfield (Pa.) and Cumberland (Md.) are also considered as standards. These coals when of good quality possess the essentials of excellence, adaptability to various kinds of furnaces, grates, boilers, and methods of firing required, besides being widely distributed and generally accessible in the Eastern market.

There are no special grades of coal mined in the Western States which are widely and generally considered as standards for testing purposes; the best coal obtainable in any particular locality being regarded as the standard of comparison.

A coal selected for maximum efficiency and capacity tests, should be the best of its class, and especially free from slagging and unusual clinker-forming impurities.

For guarantee and other tests with a specified coal containing not more than a certain amount of ash and moisture, the coal selected should not be higher in ash and in moisture than the stated amounts, because any increase is liable to reduce the efficiency and capacity more than the equivalent proportion of such increase.

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[1] This code relates primarily to tests made with coal. For reference to oil and gas fuel tests see Section 17, Page 1725.

The size of the coal, especially where it is of the anthracite class, should be determined by screening a suitable sample. (See page 1694 and Appendix No. 7 for list of sizes.)

3 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for boiler tests are:

(*a*) Platform scales for weighing coal and ashes
(*b*) Graduated scales attached to the water glasses
(*c*) Tanks and platform scales for weighing water (or water meters calibrated in place)
(*d*) Pressure gages, thermometers, and draft gages
(*e*) Calorimeters for determining the calorific value of fuel and the quality of steam
(*f*) Furnace pyrometers
(*g*) Gas analyzing apparatus

Full directions regarding the use and calibration of the above-mentioned appliances are given under the heading Apparatus and Instruments, pages 1691 to 1702, and on pages 1713 and 1714.

For particulars regarding the best location of the various instruments and apparatus, see Appendix No. 24.

4 OPERATING CONDITIONS

Determine what the operating conditions and method of firing should be to conform to the object in view, as pointed out on page 1703, and see that they prevail throughout the trial, as nearly as possible.

Where uniformity in the rate of evaporation is required, arrangement can usually be made to dispose of the steam so that this result can be attained. In a single boiler it may be accomplished by discharging steam through a waste pipe and regulating the amount by means of a valve. In a battery of boilers, in which only one is tested, the draft may be regulated on the remaining boilers to meet the varying demands for steam, leaving the test boiler to work under a steady rate of evaporation.

5 DURATION

The duration of tests to determine the efficiency of a hand-fired boiler, should be 10 hours of continuous running, or such time as may be required to burn a total of 250 lb. of coal per square foot of grate.

In the case of a boiler using a mechanical stoker, the duration, where practicable, should be at least 24 hours. If the stoker is of a type that permits the quantity and condition of the fuel bed at beginning and end of the test to be accurately estimated, the duration may be reduced to 10 hours, or such time as may be required to burn the above noted total of 250 lb. per sq. ft.

In commercial tests where the service requires continuous operation night and day, with frequent shifts of firemen, the duration of the test, whether the boilers are hand-fired or stoker-fired, should be at least 24 hours. Likewise in commercial tests, either of a single boiler or of a plant of several boilers, which operate regularly a certain number of hours and during the balance of the day the fires are banked, the duration should not be less than 24 hours.

The duration of tests to determine the maximum evaporative capacity of a boiler, without determining the efficiency, should not be less than three hours.

6 STARTING AND STOPPING

The conditions regarding the temperature of the furnace and boiler, the quantity and quality of the live coal and ash on the grates, the water level, and the steam pressure, should be as nearly as possible the same at the end as at the beginning of the test.

To secure the desired equality of conditions with hand-fired boilers, the following method should be employed:

The furnace being well heated by a preliminary run, burn the fire low, and thoroughly clean it, leaving enough live coal spread evenly over the grate (say two to four in.)[1], to serve as a foundation for the new fire. Note quickly the thickness of the coal bed as nearly as it can be estimated or measured; also the water level[2], the steam pressure, and the time, and record the latter as the starting time. Fresh coal should then be fired from that weighed for the test, the ashpit thoroughly cleaned, and the regular work of the test proceeded with.

Before the end of the test the fire should again be burned low and cleaned in such a manner as to leave the same amount of live coal on the grate as at the start. When this condition is reached, observe quickly the water level[2], the steam pressure, and the time, and record the latter as the stopping time. If the water level is not the same as at the beginning, a correction should be made by computation, rather than by feeding additional water after the final readings are taken. Finally remove the ashes and refuse from the ashpit.

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[1] 1 to 2 in. for small anthracite coals.

[2] Do not blow down the water-glass column for at least one hour before these readings are taken. An erroneous indication may otherwise be caused by a change of temperature and density of the water within the column and connecting pipe.

In a plant containing several boilers where it is not practicable to clean them simultaneously, the fires should be cleaned one after the other as rapidly as may be, and each one after cleaning charged with enough coal to maintain a thin fire in good working condition. After the last fire is cleaned and in working condition, burn all the fires low (say 4 to 6 in.), note quickly the thickness of each, also the water levels, steam pressure, and time, which last is taken as the starting time. Likewise when the time arrives for closing the test, the fires should be quickly cleaned one by one, and when this work is completed they should all be burned low the same as at the start, and the various observations made as noted.

In the case of a large boiler having several furnace doors requiring the fire to be cleaned in sections one after the other, the above directions pertaining to starting and stopping in a plant of several boilers may be followed.

To obtain the desired equality of conditions of the fire when a mechanical stoker other than a chain grate is used, the procedure should be modified where practicable as follows:

Regulate the coal feed so as to burn the fire to the low condition required for cleaning. Shut off the coal-feeding mechanism and fill the hoppers level full. Clean the ash or dump plate, note quickly the depth and condition of the coal on the grate, the water level[1], the steam pressure, and the time, and record the latter as the starting time. Then start the coal-feeding mechanism, clean the ashpit, and proceed with the regular work of the test.

When the time arrives for the close of the test, shut off the coal-feeding mechanism, fill the hoppers and burn the fire to the same low point as at the beginning. When this condition is reached, note the water level, the steam pressure, and the time, and record the latter as the stopping time. Finally clean the ash plate and haul the ashes.

In the case of chain grate stokers, the desired operating conditions should be maintained for half an hour before starting a test and for a like period before its close, the height of the throat plate and the speed of the grate being the same during both of these periods.

7 RECORDS

The records of data should be obtained as pointed out in Part I, page 1703. Half-hourly readings of the instruments are usually sufficient. If there are sudden and wide fluctuations, the readings in such cases should be taken every fifteen minutes, and in some instances oftener.

The coal should be weighed and delivered to the firemen in portions sufficient for one hour's run, thereby ascertaining the degree of uniformity of firing. An ample supply of coal should be maintained at all times, but the quantity on the floor at the end of each hour should be as small as practicable, so that the same may be readily estimated and deducted from the total weight.

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[1] See footnote, p. 1719.

The records should be such as to ascertain also the consumption of feedwater each hour, and thereby determine the degree of uniformity of evaporation.

8 QUALITY OF STEAM

If the boiler does not produce superheated steam the percentage of moisture in the steam should be determined by the use of a throttling or separating calorimeter, in the manner pointed out in Part III, page 1713. If the boiler has superheating surface, the temperature of the steam should be determined by the use of a thermometer inserted in a thermometer well, as pointed out on page 1696, Part I.

9 SAMPLING AND DRYING COAL

During the progress of the test the coal should be regularly sampled for the purpose of analysis and determination of moisture, in the manner pointed out in Part III, page 1712.

10 ASHES AND REFUSE

The ashes and refuse withdrawn from the furnace and ashpit during the progress of the test and at its close should be weighed so far as possible in a dry state. If wet the amount of moisture should be ascertained and allowed for, a sample being taken and dried for this purpose. This sample may serve also for analysis and the determination of unburned carbon and fusing temperature.

11 CALORIFIC TESTS AND ANALYSES OF COAL

The quality of the fuel should be determined by calorific tests and analysis of the coal sample above referred to. Directions for making these tests and analyses will be found in Part I under the headings Coal Calorimeters and Coal Analysis Apparatus, pages 1697 and 1698 and in Appendices Nos. 12 and 14.

12 ANALYSES OF FLUE GASES

For approximate determinations of the composition of the flue gases, the Orsat apparatus, or some modification thereof, should be employed. If momentary samples are obtained the analyses should be made as frequently as possible, say every 15 to 30 minutes, depending on the skill of the operator, noting at the time the sample is drawn the furnace and firing conditions. If the sample drawn is a continuous one, the intervals may be made longer.

Fuller directions will be found in Part I, under the heading Gas Analysis Apparatus, page 1698, and in Appendix No. 15.

13 SMOKE OBSERVATIONS

In tests of bituminous coals requiring a determination of the amount of smoke produced, observations should be made regularly throughout the trial at intervals of five minutes (or if necessary every minute), noting at the same time the furnace and firing conditions.

Full particulars regarding these observations are given in Part I, under the heading Appliances and Methods Pertaining to Smoke Determination, page 1698, and in Appendices Nos. 16 and 17.

14 CALCULATION OF RESULTS

The methods to be followed in expressing and calculating those results which are not self-evident are explained as follows:

(*a*) *Efficiency.* The "efficiency of boiler, furnace and grate" is the relation between the heat absorbed per pound of coal fired, and the calorific value of one pound of coal.

The "efficiency of boiler and furnace" is the relation between the heat absorbed per pound of combustible burned, and the calorific value of one pound of combustible. This expression of efficiency furnishes a means for comparing one boiler and furnace with another, when the losses of unburned coal due to grates, cleanings, etc., are eliminated.

The "combustible burned" is determined by subtracting from the weight of coal supplied to the boiler, the moisture in the coal, the weight of ash and unburned coal withdrawn from the furnace and ashpit, and the weight of dust, soot, and refuse, if any, withdrawn from the tubes, flues, and combustion chambers, including ash carried away in the gases, if any, determined from the analyses of coal and ash. The "combustible" used for determining the calorific value is the weight of coal less the moisture and ash found by analysis.

The "heat absorbed" per pound of coal, or combustible, is calculated by multiplying the equivalent evaporation from and at 212 deg. per lb. of coal or combustible by 970.4.

(*b*) *Corrections for Moisture in Steam.* When the percentage is less than 2 per cent it is sufficient merely to deduct the percentage from the weight of water fed. If the percentage is greater than 2 per cent or if extreme accuracy is required, the factor of correction equals

$$Q + P\frac{(T - h)}{(H - h)}$$

in which Q is the quality of the steam (one minus the decimal representing the percentage of moisture), P the proportion of moisture, T the total heat of water at the temperature of the steam, h the total heat of the feedwater, and H the total heat of saturated steam of the given temperature.

(*c*) *Correction for live steam, if any, used for aiding Combustion.* If live steam is admitted into the furnace or ash-pit for producing blast, injecting fuel, or aiding combustion, it is to be deducted from the total evaporation, and the net evaporation used in the various calculations.

(*d*) *Equivalent Evaporation.* The equivalent evaporation from and at 212 deg. is obtained by multiplying the weight of water evaporated, corrected for moisture in steam, by the "factor of evaporation." The latter equals

$$\frac{H - h}{970.4}$$

in which H and h are respectively the total heat of saturated steam and of the feedwater entering the boiler. When the steam is superheated, the total heat of the steam is that of saturated steam plus the product of the number of degrees of superheating by the specific heat of the steam.

Unless otherwise provided, a combined boiler and superheater should be treated as one unit, and the equivalent of the work done by the superheater should be included in the evaporative work of the boiler.

(*e*) *Heat Balance.* The "heat balance," or approximate distribution of the calorific value of the coal or combustible among the several items of heat utilized and heat lost, should be obtained in cases where the flue gases have been analyzed and a complete analysis made of the coal.

The loss due to moisture in the coal is found by multiplying the total heat of one pound of superheated steam at the temperature of the escaping gases, calculated from the temperature of the air in the boiler room, by the proportion of moisture.

The loss due to moisture formed by the burning of hydrogen is obtained by multiplying the total heat of one pound of superheated steam at the temperature of the escaping gases, calculated from the temperature of the air in the boiler room, by the proportion of the hydrogen, determined from the analysis of the coal, and multiplying the result by 9.

The loss due to heat carried away in the dry gases is found by multiplying the weight of gas per pound of coal or combustible by the elevation of temperature of the gases above the temperature of the boiler room, and by the specific heat of the gases (0.24). The weight of gas referred to is obtained by finding the weight of dry gas per pound of carbon burned, using the formula

$$\frac{11\,CO_2 + 8\,O + 7\,(CO + N)}{3\,(CO_2 + CO)}$$

in which CO_2, CO, O, and N are expressed in percentages by volume, and multiplying this result by the proportion borne by the carbon burned to the whole amount of coal or combustible as determined from the results of the analysis of the coal, ash, and refuse.

The loss due to incomplete combustion of carbon is found by first obtaining the proportion borne by the carbon monoxide in the gases to the sum of the carbon monoxide and carbon dioxide, and then multiplying this proportion by the proportion of carbon in the coal or combustible. and finally multiplying the product by 10,150, which is the number of heat units generated by burning to carbon dioxide one pound of carbon contained in carbon monoxide.

The loss due to combustible matter in the ash and refuse is found by multiplying the proportion that this combustible bears to the whole amount of coal or combustible, by its calorific value per pound. For most purposes it is sufficient to assume the latter to be 14,600 B.t.u., the same as that of carbon.

The loss due to moisture in the air is determined by multiplying the weight of such moisture per pound of coal or combustible by the elevation of temperature of the flue gases above the temperature of the boiler room and by 0.47. The weight of moisture is found by multiplying the weight of air per pound of coal or combustible by the moisture in one pound of air determined from readings of the wet and dry-bulb thermometer.

(*f*) *Total Heat of Combustion of Coal, by Analysis.* The total heat of combustion may be computed from the results of the ultimate analysis by using the formula

$$14.600\,C + 62.000\,(H - \frac{O}{8}) + 4000\,S,$$

in which C, H, O, and S refer to the proportions of carbon, hydrogen, oxygen, and sulphur, respectively.

(*g*) *Air for Combustion.* The quantity of air used may be calculated by the formulae:

$$\text{Lb. of air per lb. of carbon} = \frac{3.032\,N}{CO_2 + CO}$$

in which N, CO_2 and CO are the percentages of dry gas obtained by analysis, and

Lb. of air per lb. of coal = lb. air per lb. C × per cent C in the coal.

The ratio of the air supply to that theoretically required for complete combustion is $\frac{N}{N - 3.782\,O}$.

15 DATA AND RESULTS

The data and results should be reported in accordance with either the Short Form or the Complete Form printed below, adding lines for data not provided for, or omitting those not required, as may conform to the object in view.

16 CHART

In trials having for an object the determination and exposition of the complete boiler performance, the entire log of readings and data should be plotted on a chart and represented graphically. See Appendix No. 23.

17 TESTS WITH OIL AND GAS FUELS

Tests of boilers using oil or gas for fuel should accord with the rules here given, excepting as they are varied to conform to the particular characteristics of the fuel. The duration in such cases may be reduced, and the "flying" method of starting and stopping employed.

The table of data and results should contain items stating character of furnace and burner, quality and composition of oil or gas, temperature of oil, pressure of steam used for vaporizing and quantity of steam used for both vaporizing and for heating.

TABLE 1 DATA AND RESULTS OF EVAPORATIVE TEST
SHORT FORM, CODE OF 1912

(1) Test of....................boiler located at......................
to determine................, conducted by...................
(2) Kind of furnace..
(3) Grate surface.......................................sq. ft.
(4) Water-heating surface...............................sq. ft.
(5) Superheating surface.................... ....sq. ft.
(6) Date ....
(7) Duration ....hr.
(8) Kind and size of coal...................................

AVERAGE PRESSURES, TEMPERATURES, ETC.

(9) Steam pressure by gage.................................lb.
(10) Temperature of feedwater entering boiler...............deg.
(11) Temperature of escaping gases leaving boiler..........deg.
(12) Force of draft between damper and boiler...............in.
(13) Percentage of moisture in steam, or number deg. of superheating ....per cent or deg.

Total Quantities

(14) Weight of coal as fired[1] . . . . lb.
(15) Percentage of moisture in coal . . . . per cent.
(16) Total weight of dry coal consumed . . . . lb.
(17) Total ash and refuse . . . . lb.
(18) Percentage of ash and refuse in dry coal . . . . per cent.
(19) Total weight of water fed to the boiler[2] . . . . lb.
(20) Total water evaporated, corrected for moisture in steam . . . . lb.
(21) Total equivalent evaporation from and at 212 deg . . . . lb.

Hourly Quantities and Rates

(22) Dry coal consumed per hour . . . . lb.
(23) Dry coal per sq. ft. of grate surface per hour . . . . lb.
(24) Water evaporated per hour corrected for quality of steam . . . . lb.
(25) Equivalent evaporation per hour from and at 212 deg . . . . lb.
(26) Equivalent evaporation per hour from and at 212 deg. per sq. ft. of water-heating surface . . . . lb.

Capacity

(27) Evaporation per hour from and at 212 deg. (same as Line 25) . . . . lb.
(28) Boiler horsepower developed (Item 27 ÷ 34½) . . . . bl.h.p.
(29) Rated capacity, in evaporation from and at 212 deg. per hour . . . . . lb.
(30) Rated boiler horsepower . . . . bl.h.p.
(31) Percentage of rated capacity developed . . . . per cent.

Economy Results

(32) Water fed per lb. of coal fired (Item 19 ÷ Item 14) . . . . lb.
(33) Water evaporated per lb. of dry coal (Item 20 ÷ Item 16) . . . . lb.
(34) Equivalent evaporation from and at 212 deg. per lb. of dry coal (Item 21 ÷ Item 16) . . . . lb.
(35) Equivalent evaporation from and at 212 deg. per lb. of combustible [Item 21 ÷ (Item 16 — Item 17)] . . . . lb.

Efficiency

(36) Calorific value of 1 lb. of dry coal . . . . B.t.u.
(37) Calorific value of 1 lb. of combustible . . . . B.t.u.
(38) Efficiency of boiler, furnace and grate $\left[100 \times \frac{\text{Item } 34 \times 970.4}{\text{Item } 36}\right]$ per cent.
(39) Efficiency of boiler and furnace $\left[100 \times \frac{\text{Item } 35 \times 970.4}{\text{Item } 37}\right]$ . . . . per cent.

Cost of Evaporation

(40) Cost of coal per ton of . . . . lb. delivered in boiler room . . . . . . dollars
(41) Cost of coal required for evaporating 1000 lb. of water from and at 212 deg . . . . dollars

---

[1] The term "as fired" means actual condition including moisture, corrected for estimated difference in weight of coal on the grate at beginning and end.

[2] Corrected for inequality of water level and steam pressure at beginning and end.

TABLE 2 DATA AND RESULTS OF EVAPORATIVE TEST COMPLETE FORM, CODE OF 1912

(1) Test of....................boiler located at......................
to determine................, conducted by...................

DIMENSIONS, PROPORTIONS, ETC.

(2) Number and kind of boilers......................................
(3) Kind of furnace...
(4) Grate surface........width........length........area.......sq. ft.
(5) Approximate width of air spaces in grate.......................in.
(6) Proportion of air space to whole grate surface............per cent.
(7) Water-heating surface......................................sq. ft.
(8) Superheating surface.......................................sq. ft.
(9) Ratio of water-heating surface to grate surface................to 1
(10) Ratio of minimum draft area to grate surface............1 to....
(11) Date ....
(12) Duration ....hr.
(13) Kind of coal..
(14) Size of coal..

AVERAGE PRESSURES, TEMPERATURES, ETC.

(15) Steam pressure by gage.....................................lb.
(16) Barometric pressure..lb.
(17) Force of draft at dampers of individual boilers................in.
(18) Force of draft in main flue near boilers.......................in.
(19) Force of draft in main flue between economizer and chimney....in.
(20) Force of draft in furnaces.....................................in.
(21) Force of blast in ashpits......................................in.
(22) State of weather..
(23) Temperature of external air..................................deg.
(24) Temperature of fireroom......................................deg.
(25) Temperature of steam...deg.
(26) Normal temperature of saturated steam.......................deg.
(27) Temperature of feedwater entering flue heater or economizer..deg.
(28) Temperature of feedwater leaving heater or economizer and entering boilers......................................deg.
(29) Temperature of gases leaving boilers.........................deg.
(30) Temperature of gases leaving economizer....................deg.

QUALITY OF STEAM.

(31) Percentage of moisture in steam......................per cent
(32) Number of degrees of superheating.........................deg.
(33) Quality of steam (dry steam = unity).........................

TOTAL QUANTITIES.

(34) Weight of coal as fired[1]...................................lb.
(35) Percentage of moisture in coal.......................per cent.
(36) Total weight of dry coal consumed............................lb.
(37) Total ash and refuse...lb.

(38) Total combustible consumed (Line 36—Line 37)................lb.
(39) Percentage of ash and refuse in dry coal..................per cent
(40) Total weight of water fed to boiler..........................lb.
(41) Total water evaporated corrected for moisture in steam........lb.
(42) Factor of evaporation, based on temperature of water entering boilers ....
(43) Total equivalent evaporation from and at 212 deg..............lb.

Hourly Quantities and Rates.

(44) Dry coal consumed per hour..................................lb.
(45) Combustible consumed per hour...............................lb.
(46) Dry coal per sq. ft. of grate surface per hour..................lb.
(47) Water evaporated per hour, corrected for quality of steam......lb.
(48) Equivalent evaporation per hour from and at 212 deg.[3]........lb.
(49) Equivalent evaporation per hour from and at 212 deg. per sq. ft. of water-heating surface[3].................................lb.

Proximate Analysis of Coal.

(50) Fixed carbon..per cent
(51) Volatile matter...per cent
(52) Moisture ....per cent
(53) Ash ....per cent

100 per cent

(54) Sulphur, separately determined...........................per cent

Ultimate Analysis of Dry Coal.

(55) Carbon (C)..per cent
(56) Hydrogen (H)..per cent
(57) Oxygen (O)..per cent
(58) Nitrogen (N)..per cent
(59) Sulphur (S)..per cent
(60) Ash ....per cent

100 per cent

(61) Moisture in sample of coal as received.....................per cent

Analysis of Ash and Refuse.

(62) Carbon ....per cent
(63) Earthy matter...per cent
    (a) SiO_2
    (b) Al_2O_3 and Fe_2O_3
    (c) MgO
    (d) CaO
(64) Temperature of fushion of ash................................deg.

Calorific Value.

(65) Calorific value of 1 lb. of dry coal by calorimeter............B.t.u.
(66) Calorific value of 1 lb. of combustible by calorimeter.........B.t.u.
(67) Calorific value of 1 lb. of dry coal by analysis................B.t.u.
(68) Calorific value of 1 lb. of combustible by analysis............B.t.u.

CAPACITY.

(69) Evaporation per hour from and at 212 deg. (same as Line 48)....lb.
(70) Boiler horsepower developed (Line 69 ÷ 34½)............bl.h.p.
(71) Rated capacity per hour, from and at 212 deg..................lb.
(72) Rated boiler horsepower............................bl.h.p.
(73) Percentage of rated capacity developed................per cent

ECONOMY RESULTS.

(74) Water fed per lb. of coal (Item 40 ÷ Item 34)..................lb.
(75) Water evaporated per lb. of dry coal (Item 41 ÷ Item 36).......lb.
(76) Equivalent evaporation from and at 212 deg. per lb. of coal fired (Item 43 ÷ Item 34)..................................lb.
(77) Equivalent evaporation from and at 212 deg. per lb. of dry coal (Item 43 ÷ Item 36).......................................lb.
(78) Equivalent evaporation from and at 212 deg. per lb. of combustible (Item 43 ÷ Item 38...............................lb.

EFFICIENCY.

(79) Efficiency of boiler, furnace, and grate $\left[100 \times \frac{\text{Item } 77 \times 970.4}{\text{Item } 65}\right]$ per cent

(80) Efficiency of boiler and furnace $\left[100 \times \frac{\text{Item } 78 \times 970.4}{\text{Item } 66}\right]$ ..per cent

COST OF EVAPORATION.

(81) Cost of coal per ton of .... lb. delivered in boiler room....dollars
(82) Cost of coal required for evaporating 1000 lb. of water under observed conditions.................................dollars
(83) Cost of coal required for evaporating 1000 lb. of water from and at 212 deg..dollars

SMOKE DATA.

(84) Percentage of smoke as observed.......................per cent
(85) Weight of soot per hour obtained from smoke meter.............

METHODS OF FIRING.

(86) Kind of firing, whether spreading, alternate, or cooking............
(87) Average thickness of fire..
(88) Average intervals between firings for each furnace during time when fires are in normal condition...........................
(89) Average interval between times of leveling or breaking up.........

ANALYSIS OF DRY GASES BY VOLUME.

(90) Carbon dioxide (CO_2)................................per cent
(91) Oxygen (O)..per cent
(92) Carbon monoxide (CO)...................................per cent
(93) Hydrogen and hydrocarbons...............................per cent
(94) Nitrogen, by difference (N)............................per cent

100 per cent

HEAT BALANCE, BASED ON DRY COAL AND COMBUSTIBLE

	Dry Coal as Fired		Combustible Burned	
	B.t.u.	Per Cent	B.t.u.	Per Cent
(95) Heat absorbed by the boiler (Line 76 or 77x970.4)				
(96) Loss due to evaporation of moisture in coal.....				
(97) Loss due to heat carried away by steam formed by the burning of hydrogen................				
(98) Loss due to heat carried away in the dry flue gases.				
(99) Loss due to carbon monoxide..................				
(100) Loss due to combustible in ash and refuse......				
(101) Loss due to heating moisture in air........... ..				
(102) Loss due to unconsumed hydrogen and hydrocarbons, to radiation, and unaccounted for.....				
(103) Total calorific value of 1 lb. of dry coal or combustible (Lines 65 and 66)....................		100		100

---

[1] The term "as fired" means actual condition including moisture, corrected for estimated difference in weight of coal on the grate at beginning and end.

[2] Corrected for inequality of water level and of steam pressure at beginning and end.

[3] The symbol U. E. meaning Units of Evaporation, may be substituted for the expression Equivalent water evaporated into dry steam from and at 212 deg.

PART V

STEAM ENGINE CODE

RULES FOR CONDUCTING TESTS OF RECIPROCATING ENGINES

1 OBJECT AND PREPARATIONS

Determine the object, take the dimensions, note the physical conditions not only of the engine but of all parts of the plant that are concerned in the determinations, examine for leakages, install the testing appliances, etc., as pointed out in the general instructions given on pages 1689 to 1714, and prepare for the test accordingly.

2 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a simple performance test of a steam engine, in which the steam consumption is determined by feedwater measurement, are:

(*a*) Tanks and platform scales for weighing water (or water meters calibrated in place)
(*b*) Graduated scales attached to the water glasses of the boilers
(*c*) Pressure gages, vacuum gages, and thermometers
(*d*) A steam calorimeter
(*e*) A barometer
(*f*) Steam engine indicators
(*g*) A planimeter
(*h*) A tachometer or other speed measuring apparatus
(*i*) A friction brake or dynamometer

Directions regarding the use and calibration of these appliances are given on pages 1691 to 1702, and on page 1713.

The determination of the heat and steam consumption of an engine by feedwater test requires the measurement of the various supplies of water fed to the boiler; that of the water discharged by separators and drips not returned to the boiler, and that of water and steam which escapes by leakage of the boiler and piping; all of these last being deducted from the total feedwater measured.

Where a surface condenser is provided and the steam consumption is determined from the water discharged by the air pump, no such measurement of drips and leakage is required, but assurance must be had that all the steam passing into the cylinders finds its way into the condenser. If the condenser leaks the defects causing it should be remedied, or suitable correction made for the leakage.

To ascertain the consumption of heat, the various feed temperatures are taken and heat calculations made accordingly. If the conditions imposed by the particular method adopted for carrying on the test depart from the usual practice, as for example where a colder supply of feedwater is used than the ordinary supply, a preliminary or subsequent run should be made to ascertain the temperatures which obtain under the usual working conditions, and the heat measurements obtained under the test-conditions appropriately corrected for such departures.

The steam consumed by steam-driven auxiliaries which are required for the operation of the engine should be included in the total steam from which the heat consumption is calculated and the quantity of steam thus used should be determined and reported.

For fuller particulars, see description of Steam Measuring apparatus, page 1692, Part I, and Appendices Nos. 3 and 25.

3 OPERATING CONDITIONS

Determine what the operating conditions should be to conform to the object in view, and see that they prevail throughout the trial, as pointed out on page 1703, Part I.

4 DURATION

A test for heat or steam consumption, with substantially constant load, should be continued for such time as may be necessary to obtain a number of successive hourly records, during which the results are reasonably uniform. For a test involving the measurement of feedwater for this purpose, five hours is sufficient duration. Where a surface condenser is used, and the measurement is that of the water discharged by the air pump, the duration may be somewhat shorter. In this case, successive half-hourly records may be compared and the time correspondingly reduced.

When the load varies widely at different times of the day, the duration should be such as to cover the entire period of variation.

The preliminary or subsequent trial for determining the working temperatures on a heat test, where the temperatures obtained under the test conditions depart from the usual temperatures, should be of such duration as may be required to secure working results.

5 STARTING AND STOPPING

The engine and appurtenances having been set to work and thoroughly heated under the prescribed conditions of test, except in cases where the object is to obtain the performance under working conditions, note the water levels in the boilers and feed reservoir, take the time and consider this the starting time. Then begin the measurements and observations and carry them forward until the end of the period determined on. When this time arrives, the water levels and steam pressure should be brought as near as practicable to the same points as at the start. This being done, again note the time and consider it the stopping time of the test. If there are differences in the water levels, proper corrections are to be applied.

Where a surface condenser is used, the collection of water discharged by the air pump begins at the starting time, and the water is thereafter measured or weighed until the end of the test; no observations of the boilers being required.

Care should be taken in cases where the activity of combustion in the boiler furnaces affects the height of water in the gage glasses that the same conditions of fire and drafts are operating at the end as at the beginning. For this reason it is best to start and stop a test without interfering with the regularity of the operation of the feed pump, provided the latter can be regulated to run so as to supply the feedwater at a uniform rate. In some cases where the supply of feedwater is irregular, as, for example, where an injector is used of a larger capacity than is required, the supply of feedwater should be temporarily shut off.

Suitable care should be observed in noting the average height of the water in the glasses, taking sufficient time to satisfactorily judge of the full extent of the fluctuation of the water line, and thereby its mean position.

6 RECORDS

The general data should be recorded as pointed out in Part I, page 1703, under the heading Records. Half-hourly readings of the instruments are sufficient, excepting where there are wide fluctuations. A set of indicator diagrams should be obtained at intervals of 20 minutes, and at more frequent intervals if the nature of the test makes it necessary. Mark on each card the cylinder and the end on which it was taken, also the time of day. Record on one card of each set the readings of the pressure gages concerned, taken at the same time. These records should subsequently be entered on the general log, together with the areas, pressures, lengths, etc., measured from the diagrams, when these are worked up.

Further directions may be found in Part I, page 1699, under the heading Indicators, and in Appendix No. 18.

7 CALCULATION OF RESULTS

(*a*) *Dry Steam.* The quantity of dry steam consumed when there is no superheating is determined by deducting the moisture found by calorimeter test from the total amount of feedwater (the latter being corrected for leakages) or from the amount of air pump discharge, as the case may be.

When there is superheating the dry steam is found by multiplying the weight of superheated steam by the factor

$$1 + \frac{C\ (T - t)}{H - h}$$

in which

C = specific heat of superheated steam at observed pressure and temperature.

T = temperature of superheated steam.

t = temperature of saturated steam.

H = total heat of saturated steam of observed pressure.

h = total heat of feedwater.

(*b*) *Heat Consumption.* The number of heat units consumed by the engine is found by multiplying the weight of feedwater consumed, corrected for leakages, by the total heat of the steam above the working feed temperature, and multiplying the product by a factor of correction expressing the quality of the steam.

If the steam contains moisture, this factor equals

$$Q + P\ \frac{T - h}{H - h}$$

in which Q is the quality of the steam (one minus the decimal representing the percentage of moisture), P the proportion of moisture, T the total heat of water at the temperature of the steam, h the total heat of the feedwater, and H the total heat of saturated steam.

If the steam is superheated, the factor is that given above under (*a*) Dry Steam.

If there are a number of sources of feedwater supply, the corresponding heat units should be determined for each supply and the various quantities added together.

The British standard of heat consumption is based on a feedwater temperature assumed to be that of the temperature of saturated steam corresponding to the observed back pressure (whether this is above or below the atmosphere), plus the temperature due to heat derived from jacket or reheater drips. It does not include the heat consumed by any auxiliaries, except jackets and reheaters.

For fuller particulars see Appendices No. 25 and No. 26.

(*c*) *Indicated Horsepower.* In a single double-acting cylinder the indicated horsepower is found by using the formula

$$\frac{P\ L\ A\ N}{33{,}000}$$

in which P represents the average mean effective pressure in pounds per sq. in. measured from the indicator diagrams, L the length of stroke in ft., A the area of the piston less one-half the area of the piston rod, or the mean area of the rod if it passes through both cylinder heads, in sq. in., and N the number of single strokes per minute.

Where extreme accuracy is required, the power developed by each side of the piston may be determined and the results added together.

(*d*) *Brake Horsepower.* The brake horsepower is found by multiplying the net weight on the brake arm (the gross weight minus the weight when the brake is entirely free) in lb., the circumference of the circle passing through the bearing point at the end of the brake arm, in ft., and the number of revolutions of the brake shaft per minute; and dividing the product by 33,000.

(*e*) *Electrical Horsepower.* The electrical horsepower for a direct-connected generator is found by dividing the output at the bus-bar, expressed in kilowatts, by the decimal 0.746. For alternating-current systems the net output is to be used, being the total output less that consumed for excitation. See Appendix No. 21.

(*f*) *Efficiency.* The efficiency is expressed by the thermal efficiency ratio, which is found by dividing the quantity 2545 by the number of heat units consumed per h.p.-hr., either indicated or brake.

See Appendix No. 13 for reference to British practice.

(*g*) *Steam accounted for by Indicator Diagrams.* The steam accounted for, expressed in pounds per i.h.p. per hour, may readily be found by using the formula

$$\frac{13{,}750}{\text{m.e.p.}}\left[\ (C+E)\,Wc - (H+E)\ Wh\ \right]$$

in which

m.e.p = mean effective pressure.
C = proportion of stroke completed at cut-off or release
E = proportion of clearance
H = proportion of stroke uncompleted at compression
Wc = weight of 1 cu. ft. steam at cut-off or release pressure
Wh = weight of 1 cu. ft. steam at compression pressure

The points of cut-off release and compression, referred to are indicated in Fig. 1.*

In multiple expansion engines the mean effective pressure to be used in the above formula is the combined m.e.p. referred to the cylinder under consideration. In a compound engine the combined m.e.p. for the h.p. cylinder is the sum of the actual m.e.p. of the h.p. cylinder and that of the l.p. cylinder multiplied by the cylinder ratio. Likewise the combined m.e.p. for the l.p. cylinder is the sum of the actual m.e.p. of the l.p. cylinder and the m.e.p. of the h.p. cylinder divided by the cylinder ratio.

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* Reproduced from Report of Committee on Standardizing Engine Tests, Fig. 122, Trans. Am. Soc. M. E., vol. 24, p. 744.

(*h*) *Cut-Off and Ratio of Expansion.* To find the percentage of cut-off, or what may best be termed the "commercial cut-off," the following rule should be observed:

> Through the point of maximum pressure during admission draw a line parallel to the atmospheric line. Through a point on the expansion line where the cut-off is complete, draw a hyperbolic curve. The intersection of these two lines is the point of commercial cut-off, and the proportion of cut-off is found by dividing the length measured on the diagram up to this point by the total length.

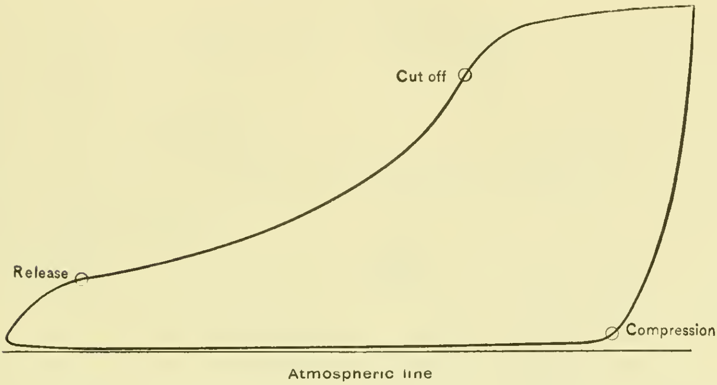

FIG. 1 POINTS WHERE "STEAM ACCOUNTED FOR BY INDICATOR" IS COMPUTED

To find the ratio of expansion divide the volume corresponding to the piston displacement, including clearance, by the volume of the steam at the commercial cut-off, including clearance.

In a multiple expansion engine the ratio of expansion is found by dividing the volume of the l.p. cylinder, including clearance, by the volume of the h.p. cylinder at the commercial cut-off, including clearance.

(*i*) *Miscellaneous.* The method of obtaining a "combined diagram" in compound and other multiple expansion engines is pointed out in Appendix No. 27. That of finding the "diagram factor" is given in Appendix No. 28. Methods of making "heat analyses," and "temperature entropy diagrams," and the manner of determining the "ratio of economy of an engine to that of an ideal engine," are given in Appendices Nos. 29, 30, and 31, respectively.

8 DATA AND RESULTS

The data and results should be reported in accordance with either the Short Form or Complete Form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view.

(See footnote on page 1744 for reference to engines driving electric generators and other machinery.)

TABLE 1 DATA AND RESULTS OF HEAT AND FEEDWATER TESTS OF STEAM ENGINE

SHORT FORM, CODE OF 1912

(1) Test of....................engine located at....................
to determine....................conducted by................
(2) Type and class of engine and auxiliaries........................

	1st Cyl.	2d Cyl.	3d Cyl.
(3) Dimensions of main engine:			
(*a*) Diameter of cylinder.........in.	......	......	......
(*b*) Stroke of piston..............ft.	......	......	......
(*c*) Diameter of piston rod each end, in	......	......	......
(*d*) Average clearance.......per cent	......	......	......
(*e*) Cylinder ratio..................	......	......	......
(*f*) Horsepower constant for 1 lb. m.e.p. and 1 r.p.m............	......	......	......

(4) Dimensions and type of auxiliaries..............................
(5) Date ....
(6) Duration ....hr.

Average Pressures and Temperatures

(7) Pressure in steam pipe near throttle by gage..................lb.
(8) Barometric pressure of atmosphere in in. of mercury..........in.
(9) Pressure in receivers by gage................................lb.
(10) Vacuum in condenser in in. of mercury........................in.
(11) Pressure in jackets and reheaters by gage.....................lb.
(12) Temperature of main supply of feedwater...................deg.
(13) Temperature of additional supplies of feedwater..............deg.

Total Quantities

(14) Total water fed to boilers from main source of supply..........lb.
(15) Total water fed from additional supplies......................lb.
(16) Total water fed to boilers from all sources....................lb.
(17) Moisture in steam or superheating near throttle....per cent or deg.
(18) Factor of correction for quality of steam.......................
(19) Total dry steam consumed for all purposes...................lb.

Hourly Quantities

(20) Water fed from main source of supply..........................lb.
(21) Water fed from additional supplies............................lb.
(22) Total water fed to boilers per hour...........................lb.
(23) Total dry steam consumed per hour.............................lb
(24) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant.................................lb.
(25) Net dry steam consumed per hour by engine and auxiliaries...lb.
(26) Net dry steam consumed per hour:
  (*a*) By engine alone...................................lb.
  (*b*) By auxiliaries....................................lb.

Heat Data

(27) Heat units per lb. of dry steam, based on temperature of Line 12...B.t.u.
(28) Heat units per lb. of dry steam, based on temperature of Line 13...B.t.u.
(29) Heat units consumed per hour, main supply of feed...........B.t.u.
(30) Heat units consumed per hour, additional supplies of feed....B.t.u.
(31) Total heat units consumed per hour for all purposes..........B.t.u.
(32) Loss of heat per hour due to leakage of plant, drips, etc......B.t.u.
(33) Net heat units consumed per hour:
  (*a*) By engine and auxiliaries..........................B.t.u.
  (*b*) By engine alone....................................B.t.u.
  (*c*) By auxiliaries.....................................B.t.u.

Indicator Diagrams

		1st Cyl.	2d Cyl.	3d Cyl.
(34)	Commercial cut-off in per cent of stroke..	......	......	......
(35)	Initial pressure in lb. per sq. in. above atmosphere	......	......	......
(36)	Back pressure at lowest point above or below atmosphere in lb. per sq. in.....	......	......	......
(37)	Mean effective pressure in lb. per sq. in...	......	......	......
(38)	Steam accounted for by indicator in lb. per i.h.p. per hour:			
	(*a*) Near cut-off....................	......	......	......
	(*b*) Near release....................	......	......	......

Speed

(39) Revolutions per minute...................................rev.
(40) Piston speed in ft. per min..................................ft.

Power

(41) Indicated horsepower developed by main-engine cylinders:
  1st cylinder..i.h.p.
  2d cylinder...i.h.p.
  3d cylinder whole engine..............................i.h.p.
  Whole engine..i.h.p.
(42) Brake horsepower..................................br.h.p.

ECONOMY RESULTS

(43) Heat units consumed by engine and auxiliaries per hour:
- (*a*) Per indicated horsepower..........................B.t.u.
- (*b*) Per brake horsepower..............................B.t.u.

(44) Dry steam consumed per indicated horsepower per hour:
- (*a*) By engine and auxiliaries..............................lb.
- (*b*) By main engine alone.................................lb.
- (*c*) By auxiliaries..lb.

(45) Dry steam consumed per brake horsepower per hour:
- (*a*) By engine and auxiliaries..............................lb.
- (*b*) By main engine alone.................................lb.
- (*c*) By auxiliaries..lb.

(46) Percentage of steam used by main-engine cylinders accounted for by indicator diagrams:
- (*a*) Near cut-off..................................per cent
- (*b*) Near release.................................per cent

SAMPLE DIAGRAMS

TABLE 2 DATA AND RESULTS OF STEAM-ENGINE TEST COMPLETE FORM, CODE OF 1912

(1) Test of....................engine located at..................
to determine...............conducted by...................

(2) Type of engine (simple, compound, or other multiple expansion; condensing or non-condensing).............................

(3) Class of engine (mill, marine, electric, etc.).....................

(4) Rated power of engine..

(5) Name of builders...

(6) Number and arrangement of cylinders of engine; how lagged; type of condenser...

(7) Type of valves...

(8) Type of boiler..

(9) Kind and type of auxiliaries (air pump, circulating pump, feed pump; jackets, heaters, etc.)................................

	1st Cyl.	2d Cyl.	3d Cyl.
(10) Dimensions of engine:			
(*a*) Single or double acting..........	......	......	......
(*b*) Cylinder dimensions:			
Bore, in......................	......	......	......
Stroke, ft....................	......	......	......
Diameter of piston rod, in......	......	......	......
Diameter of tail rod, in.......	......	......	......
(*c*) Clearance in per cent of volume displaced by piston per stroke:			
Head-end, per cent...........	......	......	......

Crank-end, per cent.......... ....
Average, per cent............. ....
(*d*) Surface in sq. ft. (average):
Barrel of cylinder, sq. ft....... ....
Cylinder heads, sq. ft.......... ....
Clearance and ports, sq. ft..... ....
Ends of piston, sq. ft.......... ....
(*c*) Jacket surfaces or internal surfaces of cylinder heated by jackets, in sq. ft.:
Barrel of cylinder, sq. ft....... ....
Cylinder heads, sq. ft.......... ....
Clearance and ports, sq. ft..... ....
Receiver jackets, sq. ft......... ....
(*g*) Horsepower constant for 1 lb m.e.p. and 1 r.p.m............ ....

(11) Dimensions of boilers:
(*a*) Number ....
(*b*) Total grate surface..............................sq. ft.
(*c*) Total water heating surface.......................sq. ft.
(*d*) Total steam heating surface.......................sq. ft.
(12) Dimensions of auxiliaries:
(*a*) Air pump..
(*b*) Circulating pump..................................
(*c*) Feed pumps..
(*d*) Heaters ....
(13) Dimensions of condenser...............................
(14) Dimensions of electric or other machinery driven by engine........
(15) Date ....
(16) Duration ....hr.

Average Pressures and Temperatures

(17) Steam pressure at boiler by gage............................lb.
(18) Steam pipe pressure near throttle, by gage...................lb.
(19) Barometric pressure of atmosphere in lb. per. sq. in............lb.
(20) Pressure in first receiver by gage............................lb.
(21) Pressure in second receiver by gage..........................lb.
(22) Vacuum in condenser:
(*a*) In in. of mercury....................................in.
(*b*) Corresponding total pressure..........................lb.
(23) Pressure in steam jacket by gage..............................lb.
(24) Pressure in reheater by gage.................................lb.
(25) Superheat in steam leaving first receiver......................deg.
(26) Superheat in steam leaving second receiver...................deg.
(27) Temperature of main supply of feedwater to boilers..........deg.
(28) Temperature of additional supplies of feedwater..............deg.
(29) Ideal feedwater temperature corresponding to the pressure of the steam in the exhaust pipe, allowance being made for heat derived from jacket or reheater drips (British Standard)....deg.

(30) Temperature of injection or circulating water entering condenser.deg.
(31) Temperature of injection or circulating water leaving condenser.deg.
(32) Temperature of air in engine room..........................deg.

Total Quantities

(33) Water fed to boilers from main source of supply.............lb.
(34) Water fed from additional supplies..........................lb.
(35) Total water fed to boilers from all sources...................lb.
(36) Moisture in steam or superheating near throttle....per cent or deg.
(37) Factor of correction for quality of steam, dry steam being unity.lb.
(38) Total dry steam consumed for all purposes.......................

Hourly Quantities

(39) Water fed from main source of supply.......................lb.
(40) Water fed from additional supplies..........................lb.
(41) Total water fed to boilers per hour.............................lb.
(42) Total dry steam consumed per hour............................lb.
(43) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant..............................lb.
(44) Net dry steam consumed per hour by engine and auxiliaries....lb.
(45) Dry steam consumed per hour:
  (*a*) Main engine alone...............................lb.
  (*b*) Jackets and reheaters...............................lb.
  (*c*) Air pump..lb.
  (*d*) Circulating pump....................................lb.
  (*e*) Feedwater pump.....................................lb.
  (*f*) Other auxiliaries.......................................lb.
(46) Injection or circulating water supplied condenser per hour....cu. ft.

Heat Data

(47) Heat units per pound of dry steam, based on temperature of Line 27..B.t.u.
(48) Heat units per pound of dry steam, based on temperature of Line 28..B.t.u.
(49) Heat units consumed per hour, main supply of feed........B.t.u.
(50) Heat units consumed per hour, additional supplies of feed....B.t.u.
(51) Total heat units consumed per hour for all purposes..........B.t.u.
(52) Loss of heat per hour due to leakage of plant, drips, etc......B.t.u.
(53) Net heat units consumed per hour:
  (*a*) By engine and auxiliaries.........................B.t.u.
  (*b*) By engine alone...............................B.t.u.
  (*c*) By auxiliaries......................................B.t.u.
(54) Heat units consumed per hour by the engine alone, reckoned from temperature given in Line 29 (British Standard)....B.t.u.

Indicator Diagrams

	1st Cyl.	2d Cyl.	3d Cyl.
(55) Commercial cut-off in per cent of stroke..	......	......	......
(56) Initial pressure in lb. per sq. in. above atmosphere	......	......	......
(57) Back-pressure at mid-stroke above or below atmosphere in lb. per sq. in.......	......	......	......

(58) Mean effective pressure in lb. per sq. in... ....

(59) Equivalent mean effective pressure in lb. per sq. in.:

(*a*) Referred to first cylinder........ ....

(*b*) Referred to second cylinder...... ....

(*c*) Referred to third cylinder....... ....

(60) Pressures and percentages used in computing the steam accounted for by the indicator diagrams, measured to points on the expansion and compression curves:

Pressure above zero in lb. per sq. in:

(*a*) Near cut-off................. ....

(*b*) Near release................ ....

(*c*) Near beginning of compression ....

Percentage of stroke at points where pressures are measured:

(*d*) Near cut-off................. ....

(*e*) Near release................ ....

(*f*) Near beginning of compression ....

(61) Aggregate m.e.p. in lb. per sq. in. referred to each cylinder given in heading...... ....

(62) Mean back pressure above zero, lb. per sq. in................................ ....

(63) Steam accounted for in lb. per indicated horsepower per hour:

(*a*) Near cut-off.................... ....

(*b*) Near release................... ....

(64) Ratio of expansion..................... ....

(65) Mean effective pressure of ideal diagram (Appendix No. 28)....lb.

(66) Diagram factor (Appendix No. 28)..............................

SPEED

(67) Revolutions per minute....................................rev.

(68) Piston speed per minute...................................ft.

(69) Variation of speed between no load and full load..............rev.

(70) Fluctuation of speed on suddenly changing from full load to no load, measured by the increase in the revolutions due to the change ....rev.

POWER

(71) Indicated horsepower developed by main engine:

1st cylinder..i.h.p.

2d cylinder...i.h.p.

3d cylinder...i.h.p.

Whole engine..i.h.p.

(72) Brake horsepower..br.h.p.

(73) Friction i.h.p. by diagrams, no load on engine, computed for average speed...i.h.p.

(74) Difference between Lines 71 and 72.........................h.p.

(75) Percentage of i.h.p. of main engine lost in friction........per cent
(76) Power developed by auxiliaries[1]..............................i.h.p.

Economy Results

(77) Heat units consumed per indicated horsepower per hour:[2]
  (*a*) By engine and auxiliaries..........................B.t.u.
  (*b*) By engine alone...................................B.t.u.
(78) Heat units consumed per brake horsepower per hour:
  (*a*) By engine and auxiliaries..........................B.t.u.
  (*b*) By engine alone...................................B.t.u.
(79) Heat units consumed by engine per hour, corresponding to ideal temperature of feedwater given in Line 29, per indicated horsepower (British Standard)..........................B.t.u.
(80) Dry steam consumed per i.h.p. per hour:
  (*a*) By engine and auxiliaries.............................lb.
  (*b*) By main engine alone.................................lb.
  (*c*) By auxiliaries.......................................lb.
(81) Dry steam consumed per brake h.p. per hour:
  (*a*) By engine and auxiliaries.............................lb.
  (*b*) By main engine alone.................................lb.
  (*c*) By auxiliaries.......................................lb.
(82) Percentage of steam used by main engine cylinders accounted for by indicator diagrams:

	1st Cyl.	2d Cyl.	3d Cyl.
(*a*) Near cut-off....................	......	......	......
(*b*) Near release....................	......	......	......

Efficiency Results

(83) Thermal efficiency ratio for engine and auxiliaries:
  (*a*) Per indicated horsepower........................per cent
  (*b*) Per brake horsepower............................per cent
(84) Thermal efficiency ratio for engine alone:
  (*a*) Per indicated horsepower........................per cent
  (*b*) Per brake horsepower............................per cent
(85) Ratio of economy of engine to that of an ideal engine working with the Rankine cycle (Appendix No. 31)..............per cent

Work Done per Heat Unit

(86) Ft.-lb. of net work per B.t.u. consumed by engine and auxiliaries (1,980,000 ÷ Line 78*a*)..................................ft.-lb.

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[1] These are not included in the power developed by the main engine.

[2] The h.p. on which the economy and efficiency results are based are those of the main engine given in Line 71.

Note: Both the Short Form and Complete Form here given refer to a steam engine used for general service.

For an engine driving an electric generator the form should be enlarged to include the electrical data, embracing the average voltage, number of amperes each phase, number of watts, number of watt hours, average power factor, etc.; and the economy results based on the electrical output embracing the heat units and steam consumed per electric h.p. per hour and per kw-hr., together with the efficiency of the generator. See table for Steam Turbine Code, Part 6.

Likewise, in a marine engine having a shaft dynamometer, the form should include the data obtained from this instrument, in which case the brake h.p. becomes the shaft h.p.

PART VI

STEAM TURBINE CODE

RULES FOR CONDUCTING TESTS OF STEAM TURBINES AND TURBO-GENERATORS

1 OBJECT AND PREPARATIONS

Determine the object, take the dimensions, note the physical conditions not only of the turbine but of the entire plant concerned, examine for leakages, install the testing appliances, etc., as pointed out in the general instructions given on pages 1689 to 1714 and prepare for the test accordingly.

2 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a simple performance test of a steam turbine or turbo-generator, in which the steam consumption is determined by feedwater measurement, are:

(*a*) Tanks and platform scales for weighing water, (or water meters calibrated in place)
(*b*) Graduated scales attached to the water glasses of the boilers
(*c*) Pressure gages, vacuum gages, and thermometers
(*d*) A steam calorimeter
(*e*) A barometer
(*f*) A tachometer or other speed-measuring apparatus
(*g*) A friction brake or dynamometer
(*h*) Volt meters, ammeters, wattmeters, and watt-hour meters for the electrical measurements in the case of a turbo-generator

Directions regarding the use and calibration of these appliances are given on pages 1691 to 1702, and on page 1713.

The determination of the heat and steam consumption of a turbine or turbo-generator should conform to the same methods as those described in the Steam Engine Code, page 1735.

The steam consumed by steam-driven auxiliaries required for the operation of a turbine should be included in the total steam from which the heat consumption is calculated the same as in the case of the steam engine.

3 OPERATING CONDITIONS

Determine what the operating conditions should be to conform to the object in view and see that they prevail throughout the trial.

4 DURATION

5 STARTING AND STOPPING

6 RECORDS

7 CALCULATION OF RESULTS

The rules pertaining to the subjects Duration, Starting and Stopping, Records, and Calculation of Results, are identically the same as those given under the respective headings in the Steam Engine Code, pages 1733 to 1737 with the single exception of the matter relating to indicator diagrams and results computed therefrom; and reference may be made to that code for the directions required in these particulars.

8 DATA AND RESULTS

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view:

DATA AND RESULTS OF STEAM TURBINE OR TURBO-GENERATOR TESTS

CODE OF 1912

(1) Test of..............turbine located at..........................
to determine..................., conducted by..................
(2) Type of turbine and class of service..............................
(3) Type of generator, kind of current, etc..........................
(4) Rated power of turbine...
(5) Type of boiler...
(6) Kind and type of auxiliaries (air pumps, circulating pumps, feed pumps, etc.)..
(7) Dimensions of turbine or turbo-generator........................
(8) Dimensions of boilers ....
(9) Dimensions of auxiliaries ....
(10) Dimensions of condenser.......................................
(11) Date ....
(12) Duration ....hr.

Average Pressures and Temperatures

(13) Steam pipe pressure near throttle, by gage.......................lb.

(14) Steam chest pressure by gage..............................lb.

(15) Barometric pressure of atmosphere in lb. per sq. in.............lb.

(16) Vacuum in condenser:

(*a*) In inches of mercury.................................in.

(*b*) Corresponding absolute pressure.....................lb.

(17) Exhaust chamber pressure (absolute).........................lb.

(18) Temperature of main supply of feedwater to boilers.............deg.

(19) Temperature of additional supplies of feedwater..............deg.

(20) Temperature of injection or circulating water entering condenser..deg.

(21) Temperature of injection or circulating water leaving condenser..deg.

Total Quantities

(22) Water fed to boilers from main source of supply.................lb.

(23) Water fed from additional supplies............................lb.

(24) Total water fed to boilers from all sources......................lb.

(25) Moisture in steam or superheating near throttle......per cent or deg.

(26) Factor of correction for quality of steam, dry steam being unity.....

(27) Total dry steam consumed for all purposes.....................lb.

Hourly Quantities.

(28) Water fed from main source of supply.........................lb.

(29) Water fed from additional supplies............................lb.

(30) Total water fed to boilers per hour...........................lb.

(31) Total dry steam consumed per hour...........................lb.

(32) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant..............................lb.

(33) Net dry steam consumed per hour.............................lb.

(34) Dry steam consumed per hour:

(*a*) By turbine...lb.

(*b*) By auxiliaries.......................................lb.

(35) Injection or circulating water supplied condensers per hour ....cu. ft.

Heat Data

(36) Heat units per pound of dry steam, based on temperature of Line 18..B.t.u.

(37) Heat units per pound of dry steam, based on temperature of Line 19..B.t.u.

(38) Heat units consumed per hour, main supply of feed...........B.t.u.

(39) Heat units consumed per hour, additional supplies of feed......B.t.u.

(40) Total heat units consumed per hour for all purposes..........B.t.u.

(41) Loss of heat per hour due to leakage of plant, drips, etc........B.t.u.

(42) Heat units consumed per hour:

(*a*) By turbine and auxiliaries...........................B.t.u.

(*b*) By turbine alone....................................B.t.u.

(*c*) By auxiliaries......................................B.t.u.

Electrical Data

(43) Average volts, each phase.................................volts

(44) Average amperes, each phase..............................amperes

(45) Average kilowatts, first meter...........................kw.

(46) Average kilowatts, second meter..........................kw.

(47) Total kilowatt output....................................kw.

(48) Power factor..

(49) Output consumed by exciter...............................kw.

(50) Net kilowatt output......................................kw.

Speed

(51) Revolutions per minute...................................rev.

(52) Variation of speed between no load and full load.............rev.

(53) Fluctuation of speed on suddenly changing from full load to no load, measured by the increase in the revolutions due to the change ....rev.

Power

(54) Brake horsepower...br.h.p.

(55) Electrical horsepower....................................h.p.

Economy Results

(56) Heat units consumed by turbine and auxiliaries per brake h.p.-hr. ....B.t.u.

(57) Dry steam consumed per brake h.p.-hr.:

  (*a*) By turbine and auxiliaries...........................lb.

  (*b*) By turbine alone.....................................lb.

  (*c*) By auxiliaries..lb.

(58) Dry steam consumed per kw-hr.:

  (*a*) By turbine and auxiliaries...........................lb.

  (*b*) By turbine alone.....................................lb.

  (*c*) By auxiliaries..lb.

Efficiency Results

(59) Thermal efficiency ratio per brake horsepower............per cent

(60) Ratio of economy of turbine to that of an ideal turbine working with the Rankine cycle..

Work Done per Heat Unit

(61) Ft.-lb. of net work per B.t.u. consumed by turbine and auxiliaries (1,980,000 ÷ Line 56)...............................ft-lb.

PART VII

CODE FOR COMPLETE STEAM POWER PLANTS

RULES FOR CONDUCTING TESTS OF COMPLETE STEAM POWER PLANTS

1 OBJECT AND PREPARATIONS

These rules are intended to apply to commercial tests of a complete plant to determine the number of pounds of fuel consumed per unit of work done in a unit of time. For tests of the component parts of a complete plant, such as boilers, engines, turbines, etc., rules may be found in the respective Codes applying to such cases.

Read the general instructions given on pages 1689 to 1704. Take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as there pointed out, and prepare for the test accordingly.

2 FUEL

Determine the character of the fuel to be used according to the object in view. For further particulars reference may be made to the Boiler Code, page 1715.

3 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a simple performance test of a steam plant are:

(*a*) Platform scales for weighing coal and ashes
(*b*) Coal calorimeter
(*c*) Steam engine indicators
(*d*) A tachometer or other speed-measuring apparatus
(*e*) Electrical instruments for determining the output of an electric plant

Directions regarding the use and calibration of these appliances are given on pages 1691 to 1702.

If the test involves the determination of boiler performance, and engine or turbine performances, additional instruments should be used as pointed out in the respective Codes referring to such tests.

4 OPERATING CONDITIONS

Determine what the operating conditions should be to conform to the object in view, and see that they prevail throughout the trial, as pointed out on page 1703.

5 DURATION

The duration of a plant test should be not less than one day of 24 hours, and preferably a full week of seven days, including Sunday.

In cases where the engine or turbine is in operation only a part of the day, the duration on which the results are computed, should be considered the length of time that the engine or turbine is in operation at its working speed.

6 STARTING AND STOPPING

In a plant operating continuously, day and night, the times fixed for starting and stopping should follow the regular periods of cleaning the fires. The fires should be quickly cleaned and then burned low, say to a thickness of 4 in. When this condition is reached the time should be noted as the starting time, and the thickness of each coal bed observed, as also the water levels and the steam pressure. Fresh coal should then be fired from that weighed for the test, the ashpits thoroughly cleaned, and the regular work of the test proceeded with. At the close of the test, following a regular cleaning, the fires should again be burned low, and when their condition has become the same as that observed at the beginning, the water levels and steam pressure also being the same, the time is observed and this time taken as the stopping time. If the water levels and steam pressure are not the same as at the beginning a suitable correction should be made by computation. The ashes and refuse are then hauled from the ashpits.

In a plant running only a part of the day, and during the balance of the day the fires are banked, the time selected for the beginning and end of the test should be that following the close of the day's run, when the fires have been burned low preparatory to cleaning and banking. The amount of live coal left on the grates under these circumstances is estimated at the beginning of the test, and the fires brought to the same condition, as near as may be, at the close of the test the next day. If the two quantities differ, a suitable correction is made in the weight of coal fired, as found by calculation.

7 RECORDS

The general data should be recorded as pointed out in Part I, page 1703, under the head of Records. Half-hourly readings of the various instruments concerned are usually sufficient, excepting where there are wide fluctuations. A set of indicator diagrams should be obtained at intervals of 20 minutes, and at more frequent intervals if the nature of the test makes it necessary. Mark on each card the cylinder and the end on which it was taken, also the time of the day. Record on one card of each set the readings of the pressure gages concerned, taken at the same time. These records should subsequently be entered on the general log, together with the areas, pressures, lengths, etc., measured from the diagrams, when these are the worked up.

Further directions may be found in Part I, page 1699, under the heading Indicators and in Appendix No. 18.

8 SAMPLING AND DRYING COAL

During the progress of the test the coal should be regularly sampled for the purpose of analysis and determination of moisture, in the manner pointed out in Part III, page 1712.

9 ASHES AND REFUSE

The ashes and refuse withdrawn from the furnace and ashpit during the progress of the test and at its close should be weighed in a dry state, and, if desired, a representative sample should be obtained for proximate analysis and the determination of the amount of unburned carbon which it contains.

10 CALORIFIC TESTS AND ANALYSES OF COAL

The quality of the fuel should be determined by calorific tests and analysis of the representative sample above referred to. Directions for making these tests and analyses will be found in Part I under the headings Coal Calorimeters and Coal Analysis Apparatus, page 1697.

11 CALCULATION OF RESULTS

The methods of calculating the indicated and electrical horsepowers are the same as explained in the Steam Engine Code, page 1736.

12 DATA AND RESULTS

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view.

DATA AND RESULTS OF COMPLETE STEAM POWER PLANT TEST

(1) Test of.................... plant located at......................
to determine................., conducted by.................
(2) Type of engine or turbine and class of service...................
(3) Rated power of engine or turbine...................................
(4) Type of boilers...
(5) Kind and type of auxiliaries (air pump, circulating pump, and feed pump; jackets, heaters, etc.).........................
(6) Dimensions of engine or turbine.....................................
(7) Dimensions of boilers ....
(8) Dimensions of auxiliaries ....
(9) Dimensions of condenser ....
(10) Date ....
(11) Duration ....hr.
(12) Length of time engine or turbine was in motion with throttle open ....hr.
(13) Length of time engine or turbine was running at normal speed...hr.
(14) Kind of coal..
(15) Size of coal..

Average Pressures and Temperatures

(16) Steam pressure at boiler by gage..................................lb.
(17) Steam pipe pressure near throttle, by gage........................lb.
(18) Barometric pressure of atmosphere in in. of mercury.............in.
(19) Pressure in receiver by gage......................................lb.
(20) Vacuum in condenser...in.
(21) Number of degrees of superheating, if any, near throttle........deg.
(22) Temperature of feedwater entering boilers.......................deg.

Total Quantities, Time, Etc.

(23) Total coal as fired[1]..lb.
(24) Moisture in coal...per cent
(25) Total dry coal consumed...lb.
(26) Ash and refuse..lb.
(27) Percentage of ash and refuse to dry coal....................per cent
(28) Calorific value by calorimeter test per lb. of dry coal.........B.t.u.
(29) Cost of coal per ton of .... lb.............................dollars

Hourly Quantities

(30) Dry coal consumed per hour, based on duration of running period ....lb.

Indicator Diagrams

(31) Mean effective pressure in lb. per sq. in.........................lb.

Electrical Data

(32) Total electrical output..kw-hr.
(33) Electrical output per hour...kw.
(34) Output consumed by exciter...kw.

Where an independent superheater is used, this includes coal burned in the superheater.

(35) Net electrical output per hour.................................kw.
(36) Average volts each phase................................volts
(37) Average amperes each phase...........................amperes
(38) Power factor..

SPEED

(39) Revolutions per minute.......................................rev.

POWER

(40) Indicated horsepower developed by main engine:
  First cylinder...i.h.p.
  Second cylinder...i.h.p.
  Whole engine..i.h.p.
(41) Net electrical horsepower.......................................h.p.

ECONOMY RESULTS

(42) Dry coal consumed per i.h.p. per hour ....lb.
(43) Dry coal consumed per kw-hr. ....lb.
(44) Cost of coal per i.h.p. per hour..............................cents
(45) Cost of coal per kw-hr. ....cents

PART VIII

PUMPING MACHINERY CODE

RULES FOR CONDUCTING DUTY TRIALS OF STEAM PUMPING MACHINERY[1]

1 OBJECT AND PREPARATIONS

Read the general instructions given on pages 1689 to 1714. Determine the object, take the dimensions, note the physical conditions not only of the pumping machinery but of all parts of the plant concerned, examine for leakages, install the testing appliances, etc., as there pointed out, and prepare for the test accordingly.

In a reciprocating pump, determine the quantity of water leakage or slip past the plungers, and that of the pump valves, if any, as pointed out in Appendix No. 32.

2 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a simple duty trial of pumping machinery, in which the steam consumption is determined by feedwater measurement, are:

(*a*) Tanks and platform scales for weighing water (or water meters calibrated in place)
(*b*) Graduated scales attached to the water glasses of the boilers
(*c*) Pressure gages, vacuum gages, and thermometers
(*d*) A steam calorimeter
(*e*) A barometer
(*f*) A tachometer or other speed-measuring apparatus
(*g*) For rotary pumps a weir or other means for measuring the quantity of water pumped
(*h*) Stroke scales, for direct-acting pumps

In trials of a reciprocating pumping engine, involving the determination of the complete performance, the weir or other means of measurement noted should be provided, and in addition the following:

(*i*) Steam engine indicators
(*j*) A planimeter

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[1] In the case of a pump driven by some other prime mover than a steam engine or steam turbine, the code may be modified to suit the particular circumstances.

Directions regarding the use and calibration of these appliances are given on pages 1691 to 1702 and page 1713.

The determination of the heat and steam consumption should conform to the same methods as those described in the Steam Engine Code, pages 1732 and 1733.

The steam consumed by steam-driven auxiliaries which are required in the operation of the pumping machinery should be included in the total steam from which the heat consumption is calculated, the same as noted in the Steam Engine Code.

3 OPERATING CONDITIONS

Determine what the operating conditions should be to conform to the object in view and see that they prevail throughout the trial, as pointed out on page 1703.

In trials for maximum duty, care should be taken that no air is snifted into the pump cylinders, causing imperfect filling. In such cases, and indeed in all cases where air is thus admitted in sufficient quantity to affect the performance as revealed by indicator diagrams from the water end, the result should be corrected accordingly.

4 DURATION

5 STARTING AND STOPPING

6 RECORDS

The rules pertaining to the subjects Duration, Starting and Stopping, and Records, are identically the same as those given under the respective headings in the Steam Engine Code, pages 46, 47 and 48; and reference may be made to that code for the necessary directions in these particulars. Where the pump-end is of the reciprocating class, the indicator diagrams should be taken not only from the steam cylinders but also from the water cylinders.

7 CALCULATION OF RESULTS

The rules pertaining to Dry Steam, Heat Consumption, and Indicated Horsepower, are identically the same as those given on pages 1735 and 1736 of the Steam Engine Code; and reference may be made to that code for the necessary directions in these particulars.

(*a*) *Water Horsepower.* The water horsepower in a reciprocating pump is found by multiplying the net area of the plunger in sq. in.

by the total head, which is made up of the pressure shown by the gage on the force main, that on the suction main, and that representing the vertical distance between the centers of the two gages, all expressed in lb. per sq. in.; the length of the stroke in ft.; and the number of single strokes per minute; and dividing the final product by 33,000.

In a rotary pump the water horsepower is found by multiplying the weight of water discharged per hour in lb., as determined by weir or other measurement; by the total head in ft., as determined from the readings of the gage on the force main, that on the suction main, and the vertical distance between the two gages; and the product divided by 1,980,000.[1]

(*b*) *Duty.* The duty per million heat units is found by dividing the number of ft-lb. of work done during the trial by the total number of heat units consumed; and multiplying the quotient by 1,000,000. The amount of work is found in the case of reciprocating pumps by multiplying the net area of the plunger in sq. in., the total head expressed in lb. per sq. in. (which is made up of the pressure shown by the gage on the force main, that on the suction main, and the vertical distance between the centers of the two gages, all reduced to lb.), by the length of the stroke in ft., and the total number of single strokes during the trial; finally correcting for the percentage of leakage of the pump. In a rotary pump the work done is found by multiplying the weight of water discharged during the trial, as determined by weir or other measurement, by the total head in ft.

The duty per 1000 lb. of dry steam is found by dividing the ft-lb. of work done, as noted above, by the total weight of dry steam, and multiplying the quotient by 1000.

(*c*) *Capacity.* The capacity in gal. per 24 hr. for reciprocating pumps is found by multiplying the net area of the plunger by the length of the stroke in ft., (in direct-connected engines the average length of stroke); then by the number of single strokes per minute; and the product of these three by the constant 74.8; finally correcting for the percentage of leakage of the pump.

(*d*) *Leakage of Pump.* The percentage of leakage is the percentage borne by the quantity of leakage found on the leakage trial, to the quantity of water discharged on the duty run determined from plunger displacement.

(*e*) *Friction.* The percentage of total friction in a reciprocating pump is the percentage borne by the friction horsepower to the indicated horsepower of the steam cylinders.

(*f*) *Miscellaneous.* For the calculation of other results pertaining specially to the performance of the steam end of a reciprocating pump, reference may be made to the Steam Engine Code.

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[1] If there is a material difference in velocity of the water at the points where the gages are attached, a correction should be made for the corresponding difference in "velocity head."

8 DATA AND RESULTS

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view.

In the case of a pumping engine of the reciprocating class for which a record of the complete performance is desired, the additional engine data and results given in the Complete Form of the Steam Engine Code may supplement those here given.

DATA AND RESULTS OF DUTY TRIAL OF STEAM PUMPING MACHINERY

CODE OF 1912

(1) Test of.....................pump, located at...................
to determine................., conducted by...................
(2) Type of machinery..
(3) Rated capacity in gal. per 24 hr...............................
(4) Type of boiler...
(5) Type of auxiliaries..
(6) Dimensions of engine or turbine...............................
(7) Dimensions of pump...
(8) Dimensions of boilers..
(9) Dimensions of auxiliaries......................................
(10) Dimensions of condenser.......................................
(11) Date ....
(12) Duration ....hr.

Average Pressures and Teperatures

(13) Steam pressure at boiler by gage....................lb. per sq. in.
(14) Steam pipe pressure near throttle, by gage............lb. per sq. in.
(15) Barometric pressure of atmosphere in in. of mercury..........in.
(16) Pressure in receiver by gage........................lb. per sq. in.
(17) Vacuum in condenser in in. of mercury...........................in.
(18) Pressure in force main by gage..................................lb.
(19) Pressure in suction main by gage................................lb.
(20) Vertical distance between centers of two gages..................ft.
(21) Total head, expressed in lb. pressure...........................lb.
(22) Total head, expressed in ft.....................................ft.
(23) Temperature of main supply of feedwater to boilers............deg.
(24) Temperature of additional supplies of feedwater...............deg.
(25) Temperature of air in engine room.............................deg.

Total Quantities

(26) Water fed to boilers from main source of supply.................lb.
(27) Water fed from additional supplies..............................lb.
(28) Total water fed to boilers from all sources.....................lb.

(29) Moisture in steam or superheating near throttle......per cent or deg.
(30) Factor of correction for quality of steam, dry steam being unity.....
(31) Total dry steam consumed for all purposes......................lb.
(32) Total leakage of plungers and valves..........................gal.
(33) Total number of gallons of water discharged:
    (*a*) By plunger displacement, uncorrected..................gal.
    (*b*) By plunger displacement, corrected for leakage.........gal.
    (*c*) By weir or other measurement of discharge............gal.
(34) Total weight of water discharged:
    (*a*) By plunger displacement, corrected......................lb.
    (*b*) By measurement of discharge..........................lb.

Hourly Quantities

(35) Water fed from main source of supply.........................lb.
(36) Water fed from additional supplies..........................lb.
(37) Total water fed to boilers per hour............................lb.
(38) Total dry steam consumed per hour............................lb.
(39) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant...............................lb.
(40) Net dry steam consumed per hour............................lb.
(41) Dry steam consumed per hour:
    (*a*) By main engine or turbine............................lb.
    (*b*) By auxiliaries...lb.
(42) Water discharged per hour:
    (*a*) By plunger displacement, corrected......................lb.
    (*b*) By measurement of discharge..........................lb.

Heat Data

(43) Heat units per lb. of dry steam based on temperature, Line 23..B.t.u.
(44) Heat units per lb. of dry steam based on temperature, Line 24..B.t.u.
(45) Heat units consumed per hour based on main supply of feed..B.t.u.
(46) Heat units consumed per hour based on additional supplies of feed ....B.t.u.
(47) Total heat units consumed per hour for all purposes........B.t.u.
(48) Loss of heat per hour due to leakage of plant, drips, etc......B.t.u.
(49) Net heat units consumed per hour..........................B.t.u.
(50) Heat units consumed per hour:
    (*a*) By engine or turbine alone..........................B.t.u.
    (*b*) By auxiliaries.....................................B.t.u.

Indicator Diagrams

(51) Mean effective pressure....................................lb.

Speed and Stroke

(52) Revolutions per minute.....................................rev.
(53) Number of single strokes per minute........................strokes
(54) Average length of stroke......................................ft.

Power

(55) Indicated horsepower developed:
    1st cylinder..i.h.p.
    2d cylinder...i.h.p.
    Whole engine..i.h.p.

(56) Water horsepower..h.p.
(57) Friction h.p. (Line 55—Line 56)........................fr. h.p.
(58) Percentage of i.h.p. lost in friction......................per cent

DUTY

(59) Duty per million heat units..................................ft.-lb.
(60) Duty per thousand lb. of steam..............................ft.-lb.

WORK DONE PER HEAT UNIT

(61) Ft.-lb. of work per B.t.u. (Line 59 ÷ 1,000,000)..............ft.-lb.

CAPACITY

(62) Number of gal. of water pumped in 24 hr.:
  (*a*) By plunger displacement, corrected for leakage........gal.
  (*b*) By weir or other measurement........................gal.
(63) Gallons of water per minute:
  (*a*) By plunger displacement, corrected for leakage........gal.
  (*b*) By weir or other measurement........................gal.

ECONOMY RESULTS, STEAM END

(64) Heat units consumed per i.h.p. per hour:[1]
  (*a*) By engine and auxiliaries........................B.t.u.
  (*b*) By engine alone..................................B.t.u.
  (*c*) By auxiliaries...................................B.t.u.
(65) Dry steam consumed per i.h.p. per hour:
  (*a*) By engine and auxiliaries..........................lb.
  (*b*) By engine alone....................................lb.
  (*c*) By auxiliaries.....................................lb.

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[1] The i.h.p. on which the economy results are based is that of the main engine given in Line 54.

PART IX

CODE FOR COMPLETE PUMPING MACHINERY PLANTS

RULES FOR CONDUCTING TESTS OF COMPLETE STEAM PUMPING MACHINERY PLANTS

1 OBJECT AND PREPARATIONS

This code applies to a commercial test of a complete steam pumping machinery plant, having for an object the determination of the fuel cost of pumping a given quantity of water for a day's run of 24 hours. For tests of the component parts of the plant, rules may be found in the respective codes applying thereto.

Read the general instructions given on pages 1689 to 1704. Take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as there pointed out and prepare for the test accordingly.

In a reciprocating pump, ascertain the quantity of water leakage or slip past the plungers, and that of the pump valves, if any, as pointed out in Appendix No. 32.

2 FUEL

Determine the character of the fuel to conform with the object in view. To obtain maximum economy or capacity, the fuel should be some kind of coal that is regarded as a standard, as noted in the Boiler Code, page 1715.

3 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a simple performance test of a complete pumping machinery plant are:

(*a*) Platform scales for weighing coal and ashes
(*b*) A coal calorimeter
(*c*) A tachometer or other speed measuring apparatus.
(*d*) A pressure gage attached to the force main
(*e*) For rotary pumps, a weir or other means for measuring the quantity of water pumped
(*f*) Stroke scales, for direct-acting pumps.

Directions regarding the use and calibration of these appliances may be found on pages 1691 to 1702.

If the test also involves the determination of boiler performance and pumping machinery performance, additional apparatus and instruments are required, as pointed out in the respective codes referring thereto.

4 OPERATING CONDITIONS

Determine what the operating conditions should be to conform to the object in view, and see that they prevail throughout the trial, as explained on page 1703, Part I.

5 DURATION

The duration of a test of a complete pumping machinery plant should be not less than one day of 24 hours.

In cases where the machinery is in operation only a part of the calendar day, the coal consumption used in computing the results should be that of the entire 24 hours.

6 MISCELLANEOUS

The methods of starting and stopping a test of a complete pumping plant, sampling and drying coal, determining ashes and refuse, and ascertaining the calorific value and chemical analysis of the coal, are the same as those described in the Code for Complete Steam Power Plants, pages 1753 and 1754.

7 CALCULATION OF RESULTS

For the calculation of water horsepower, duty, capacity, and leakage of pump, reference may be made to the explanations given on pages 1759 and 1760 of the Pumping Engine Code.

8 RECORDS, DATA, AND RESULTS

The records should be made as explained on page 1703, and the tabular summary of data and results reported in accordance with the form given herewith, adding lines not provided for, or omitting those not required.

DATA AND RESULTS OF TEST OF STEAM PUMPING MACHINERY PLANT

CODE OF 1912

(1) Test of....................plant located at....................
to determine.............conducted by....................
(2) Type of machinery..
(3) Rated capacity in gallons per 24 hrs..............................
(4) Type of boilers ....
(5) Type of auxiliaries ....
(6) Dimensions of machinery ....
(7) Dimensions of boilers ....
(8) Dimensions of auxiliaries ....
(9) Dimensions of condenser ....
(10) Date ....
(11) Duration ....hr.
(12) Length of time machinery was in motion with throttle open....hr.
(13) Length of time machinery was running at normal speed........hr.

Average Pressures and Temperatures

(14) Steam pressure at boiler by gage..................lb. per sq. in.
(15) Pressure in force main by gage............................lb.
(16) Pressure in suction main by gage..........................in.
(17) Vertical distance between centers of two gages................ft.
(18) Total head expressed in lb. pressure..........................lb.
(19) Total head expressed in ft....................................ft.

Speed and Stroke

(20) Revolutions per minute....................................rev.
(21) Number of single strokes per minute..................strokes
(22) Average length of stroke..................................ft.

Total Quantities.

(23) Total coal as fired.......................................lb.
(24) Moisture in coal.....................................per cent
(25) Total dry coal consumed...................................lb.
(26) Ash and refuse..lb.
(27) Percentage of ash and refuse to dry coal..............per cent
(28) Calorific value per lb. of dry coal by calorimeter test........B.t.u.
(29) Cost of coal per ton of ...... lb........................dollars
(30) Total leakage of plungers and valves......................gal.
(31) Total number of gals of water discharged:
 (*a*) By plunger displacement, uncorrected.................gal.
 (*b*) By plunger displacement, corrected for leakage.........gal.
 (*c*) By weir or other measurement.........................gal.

Hourly Quantities

(32) Dry coal consumed per hour, based on duration of running period, (Line 25÷Line 12)..............................lb.

Power

(33) Water horsepower..h.p.

Economy Results

(34) Dry coal consumed per water h.p. per hour..................lb.

(35) Cost of coal per water h.p. per hour......................cents

(36) Duty per 100 lb. of dry coal...................................

Capacity

(37) Number of gal. of water pumped in 24 hr.:
   (*a*) By plunger displacement, corrected for leakage........gal.
   (*b*) By weir or other measurement.......................gal.

(38) Number of gal. of water pumped per minute:
   (*a*) By plunger displacement, corrected for leakage........gal.
   (b) By weir or other measurement.......................gal.

PART X

CODE FOR AIR MACHINERY

RULES FOR CONDUCTING TESTS OF STEAM-DRIVEN COMPRESSORS, BLOWERS AND FANS[1]

1 OBJECT AND PREPARATIONS

2 APPARATUS AND INSTRUMENTS

3 OPERATING CONDITIONS

4 DURATION

5 STARTING AND STOPPING

6 RECORDS

The directions pertaining to the above divisions of the subject are almost identical with those given in the Pumping Machinery Code under the same headings, the only difference being that the work done by the steam end is expended in moving a volume of air, instead of a body of water.

The quantity of air discharged should be measured by a gasometer, or by delivery into tanks of known capacity. Where these means are not available the pitot tube should be used, as described in Appendix No. 6, or some other means should be employed which is subject to calibration.

If the air end is of the reciprocating type, indicator diagrams should be regularly taken from this end as well as from the steam end.

7 CALCULATION OF RESULTS

The rules pertaining to dry steam, heat consumption, and indicated horsepower of the steam end, are identically the same as those given on pages 1735 and 1736 of the Steam Engine Code, and reference may be made to that code for the necessary directions in these particulars.

(*a*) *Air Horsepower.* The gross work done at the air end of a reciprocating machine, expressed in horsepower, is found by multiplying together the net area of the air piston in sq. in., the mean effective air pressure in lb. per sq. in. as determined from indicator diagrams, the length of the stroke in ft., and the number of single strokes per minute; and dividing their product by 33,000.

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[1] In the case of air machinery driven by some other prime mover than a steam engine or turbine, the code may be modified to meet the particular requirements.

The net work at the air end of either reciprocating or rotary machines, expressed in ft-lb. per minute, is found by multiplying the corrected volume of the compressed air in cu. ft. discharged into the main delivery pipe per minute, by the impact or total pressure in lb. per sq. ft. and by the hyperbolic logarithm of the ratio of the total pressure to the atmospheric pressure (all pressures being absolute pressures). The net air horsepower is found by dividing the product by 33,000. The corrected volume of the compressed air may be found by multiplying the sectional area of the delivery main in sq. ft. by the mean velocity in ft. per minute as determined by pitot tube or other measurement, and reducing the result to atmospheric temperature by multiplying by the proportion.

$$\frac{460 + t}{460 + T}$$

in which t is the temperature of the air supplied to the machine and T the temperature of the air in the delivery main.

(*b*) *Capacity.* The capacity is the number of cu. ft. of air discharged through the delivery main per minute, as determined by gasometer, tank or other mode of measurement, reduced to the equivalent free air at the atmospheric temperature and pressure. The correction for pressure is made by multiplying by the proportion $\frac{P_2}{P_1}$, in which P_1 is the atmospheric pressure and P_2 the total pressure in the main (absolute pressures), and the correction for temperature as above.

The capacity may also be expressed in the number of cu. ft. of compressed air discharged per minute at a given pressure above the atmosphere reduced to the atmospheric temperature.

(*c*) *Miscellaneous.* For methods of calculating results pertaining especially to the performance of the steam-end of a reciprocating air pumping machine, reference may be made to the Steam Engine Code, pages 1735, 1736, and 1737.

The "efficiency of compression" in a reciprocating machine is determined by first ascertaining the net work at the air end given above under the heading, "(*a*) Air Horsepower," and then dividing the net work thus found by the gross work given under the same heading.

The "mechanical efficiency" of a reciprocating machine is determined by dividing the gross air horsepower at the air end by the indicated horsepower at the steam end, or by the horsepower delivered by the belt or motor in the case of other means of driving.

8 DATA AND RESULTS

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for and omitting those not required, as may conform with the object in view.

In the case of an air-pumping machine of the reciprocating class for which a record of the complete performance is desired, the additional engine data and results given in the Complete Form of the Steam Engine Code may supplement those here given:

DATA AND RESULTS OF TEST OF AIR MACHINERY

CODE OF 1912

(1) Test of.........................located at...................
to determine...................conducted by...............
(2) Type of machinery...
(3) Rated capacity in cu. ft. of free air per minute..................
(4) Rated capacity in cu. ft. of air discharged per minute at 100 lb. per sq. in. above atmosphere, reduced to the atmospheric temperature ....cu. ft.
(5) Type of boilers ....
(6) Type of auxiliaries ....
(7) Dimensions of engine or turbine at steam end...................
(8) Dimensions of cylinders or blowers at air end..................
(9) Dimensions of boilers ....
(10) Dimensions of auxiliaries ....
(11) Dimensions of condenser ....
(12) Date ....
(13) Duration ....hr.

Average Pressures and Temperatures

(14) Steam pressure at boiler by gage...................lb. per sq. in.
(15) Steam pipe pressure near throttle, by gage..........lb. per sq. in.
(16) Barometric pressure of atmosphere in in. of mercury..........in.
(17) Pressure in receiver by gage.......................lb. per sq. in.
(18) Vacuum in condenser in in. of mercury......................in.
(19) Pressure in delivery main by gage (impact pressure)..........lb.
(20) Total head, expressed in ft.................................ft.
(21) Temperature of main supply of feedwater to boilers............deg.
(22) Temperature of additional supplies of feedwater..............deg.
(23) Temperature of air in engine room or air supplied to machine..deg.
(24) Temperature by wet-bulb thermometer.......................deg.
(25) Temperature of air in delivery main........................deg.

Total Quantities

(26) Water fed to boilers from main source of supply..............lb.
(27) Water fed from additional supplies...........................lb.
(28) Total water fed to boilers from all sources...................lb.
(29) Moisture in steam or superheating near throttle....per cent or deg.
(30) Factor of correction for quality of steam, dry steam being unity...
(31) Total dry steam consumed for all purposes...................lb.
(32) Total cu. ft. of compressed air delivered as measured........cu. ft.

(33) Total cu. ft. of compressed air delivered reduced to atmospheric temperature and pressure..............................lb.

(34) Total weight of air delivered..............................lb.

HOURLY QUANTITIES

(35) Water fed from main source of supply..........................lb.

(36) Water fed from additional supplies..........................lb.

(37) Total water fed to boilers per hour..........................lb.

(38) Total dry steam consumed per hour..........................lb.

(39) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant..............................lb.

(40) Net dry steam consumed per hour..........................lb.

(41) Dry steam consumed per hour:

  (*a*) By main engine or turbine..........................lb.

  (*b*) By auxiliaries ....lb.

(42) Cu. ft. of compressed air delivered per hour as measured......cu. ft.

(43) Cu. ft. of compressed air delivered per hour reduced to atmospheric temperature....................................cu. ft.

(44) Cu. ft. of compressed air delivered per hour reduced to atmospheric temperature and pressure..............................cu. ft.

(45) Weight of air delivered per hour..........................lb.

HEAT DATA

(46) Heat units per lb. of dry steam based on temperature Line 21..B.t.u.

(47) Heat units per lb. of dry steam based on temperature Line 22..B.t.u.

(48) Heat units consumed per hour based on main supply of feed...B.t.u.

(49) Heat units consumed per hour based on additional supplies of feed ....B.t.u.

(50) Total heat units consumed per hour for all purposes.........B.t.u.

(51) Loss of heat per hour due to leakage of plant, drips, etc....B.t.u.

(52) Net heat units consumed per hour............................B.t.u.

(53) Heat units consumed per hour:

  (*a*) By engine or turbine alone..........................B.t.u.

  (*b*) By auxiliaries......................................B.t.u.

INDICATOR DIAGRAMS

(54) Mean effective pressure in steam cylinders......................lb.

(55) Mean effective pressure in air cylinders......................lb.

SPEED AND STROKE

(56) Revolutions per minute....................................rev.

(57) Number of single strokes per minute......................strokes

POWER

(58) Indicated horsepower developed at steam end of reciprocating machine ....i.h.p.

(59) Gross air horsepower as indicated in air cylinders of reciprocating machine ....air h.p.

(60) Net air horsepower as computed from Line 43............air h.p.

(61) Friction of reciprocating machine (Line 58 — Line 59)....fr. h.p.

(62) Percentage of i.h.p. lost in friction of machine..........per cent

ECONOMY RESULTS, STEAM END OF ENGINE-DRIVEN MACHINES

(63) Heat units consumed per i.h.p. per hour:
  (*a*) By engine and auxiliaries........................B.t.u.
  (*b*) By engine alone........................B.t.u.
  (*c*) By auxiliaries........................B.t.u.
(64) Dry steam consumed per i.h.p. per hour[1]:
  (*a*) By engine and auxiliaries........................lb.
  (*b*) By engine alone........................lb.
  (*c*) By auxiliaries........................lb.

ECONOMY RESULTS, AIR DELIVERED

(65) Heat units consumed per hour per net air h.p. of Line 60:
  (*a*) By engine or turbine and auxiliaries..............B.t.u.
  (*b*) By engine or turbine alone........................B.t.u.
  (*c*) By auxiliaries........................B.t.u.
(66) Dry steam consumed per hour per net h.p. of Line 60:
  (*a*) By engine or turbine and auxiliaries..................lb.
  (*b*) By engine or turbine alone........................lb.
  (*c*) By auxiliaries........................lb.

EFFICIENCY RESULTS

(67) Thermal efficiency ratio for engine alone:
  (*a*) Per i.h.p., steam end (2545 ÷ Line 63*b*)..........per cent
  (*b*) Per net air h.p.. air delivery (2545 ÷ Line 65*b*)....per cent

WORK DONE PER HEAT UNIT

(68) Ft.-lb. of net work per B.t.u. consumed by engine or turbine and auxiliaries (1,980,000 ÷ Line 65*a*)......................ft-lb.

CAPACITY

(69) Cu. ft. of compressed air delivered per minute as measured....cu. ft.
(70) Cu. ft. of compressed air delivered per minute, reduced to atmospheric temperature........................cu. ft.
(71) Cu. ft. of compressed air delivered per minute at 100 lb. pressure, reduced to atmospheric temperature................cu. ft.
(72) Cu. ft. of compressed air delivered per minute reduced to atmospheric temperature and pressure (free air)..............cu. ft.

MISCELLANEOUS RESULTS

STEAM DRIVEN RECIPROCATING MACHINE

(73) Efficiency of compression (Line 60 ÷ Line 59 × 100)......per cent
(74) Mechanical efficiency of machine (Line 59 ÷ Line 58 × 100).per cent
(75) Volumetric efficiency (Line 72 ÷ 1st Compr. Displ. × 100)...per cent

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[1] The i.h.p. on which these economy results are based is that of the main engine given in Line 58.

Note: In the case of air compressors having more than one stage and in those having intercoolers, additional data should be given covering pressures and temperatures in the different stages, the quantity of water used for cooling and temperatures of the air and water entering and leaving the cooler.

PART XI

LOCOMOTIVE CODES

RULES FOR CONDUCTING TESTS OF LOCOMOTIVES

CODE FOR SHOP TESTS

1 INTRODUCTION

Locomotive tests are of two leading kinds, shop tests and road tests. The former are made under conditions quite similar to those of a stationary testing plant in which the power is absorbed by a brake. The latter are made under conditions of service on the road, the locomotive hauling a train of cars. Being a complete steam plant in itself, embracing boiler, engine, and certain auxiliaries, a locomotive equipped for a shop test should be considered in the same class as a stationary plant, and the code of rules for such tests are therefore similar to those given in Part VII.

2 OBJECT AND PREPARATIONS

The leading object of a shop test, so far as covered by this code, is the determination of the coal and steam consumption per unit of power when the locomotive is operated under fixed conditions. There are other less prominent objects, such as a determination of the performance of the engine alone under different conditions, or of the boiler running under various rates of combustion, but tests having these objects are substantially covered by the Boiler and Engine Codes of Parts IV and V, to which reference may be made.

Note the general instructions given on pages 1689 to 1714 so far as they pertain to the work in hand. Take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as there pointed out, and make preparations for the test accordingly.

It is assumed that a testing plant is available where the driving wheels can be mounted upon friction brake apparatus for suitably disposing of and measuring the power given off, as further explained in Appendix No. 33.

3 FUEL[1]

Select the fuel required in accordance with the special object in view. If maximum efficiency or capacity is desired the fuel should be some kind of coal that is regarded as a standard for the locality where the test is made, as noted in the Boiler Code, page 1715.

4 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a shop test of a locomotive are

(*a*) Platform scales for weighing coal and ashes
(*b*) A coal calorimeter
(*c*) Tanks and scales for weighing water (or water meters calibrated in place)
(*d*) A graduated scale attached to the water glass
(*e*) Pressure gages, draft gages, and thermometers
(*f*) A steam calorimeter
(*g*) Steam engine indicators
(*h*) A planimeter
(*i*) A tachometer or other speed-measuring apparatus
(*j*) Friction brake apparatus for absorbing and measuring the power given off
(*k*) A dynamometer for determining the pull on the drawbar

Directions regarding their use and calibration are given on pages 1691 to 1702, page 1713, and Appendix No. 34.

5 OPERATING CONDITIONS

Determine what the operating conditions and method of firing should be as explained on page 1703 and see that they prevail throughout the trial.

6 DURATION

The duration of a simple shop test of a locomotive depends somewhat upon the rate of combustion, but it should not be less than two hours. In any case it should continue until at least 250 lb. of coal have been burned on each square foot of grate.

On account of liability of error in estimating the difference in the amount of coal in the furnace at beginning and end, the results of a single test should not be relied upon, but repeated tests should be made under the required conditions and the results averaged.

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[1] See Section 14, Page 1781, for memoranda relating to oil fuel.

7 STARTING AND STOPPING

Set the engine to work with a comparatively light load at the normal speed. Burn the fire down as low as practicable and clean it of ashes and clinkers by rocking the grate, afterwards hauling the ash-pan. Note the average thickness of the firebed, quickly observe the steam pressure, water level, and time, and record the latter as the starting time. Immediately fire with weighed coal and increase the load as fast as may be to the working point, meanwhile establishing the furnace conditions determined on. Then proceed with the regular work of the test.

When the end of the test approaches, again burn down and clean the fire, meanwhile reducing the load as may be needed to prevent undue force of draft. As soon as the firebed reaches the condition observed at the start, immediately take the steam pressure, water level, and time, and record the latter as the stopping time. Finally haul the ashes and refuse from the ash-pan and remove the cinders from the smoke-box. If the firebed or water level at the end differs from that observed at the beginning, a correction is applied accordingly.

8 RECORDS

The general data should be recorded as pointed out in Part I, page 1703. Readings of the instruments concerned and a set of indicator diagrams should be obtained every ten minutes. A suitable number of additional diagrams should be taken during the operations of starting and stopping to obtain the average load for these short periods. Directions on the subject of indicating may be found on pages 1699 and 1736 and in Appendix No. 18.

9 SAMPLING AND DRYING COAL

During the progress of the trial the coal should be regularly sampled and its moisture determined in the manner pointed out on page 1712.

10 ASHES AND REFUSE

The ashes and refuse withdrawn from the ash-pan and smoke-box should be weighed in a dry state, and sampl s should be taken and analyzed if desired for unburned carbon.

11 CALORIFIC TESTS OF COAL

The quality of the coal should be determined by a calorimeter test of the sample above referred to. Methods of making this test are described on page 1698 and in Appendix No. 12.

12 CALCULATION OF RESULTS

For methods of calculating the results, reference may be made to page 1722 of the Boiler Code and page 1735 of the Engine Code.

13 DATA AND RESULTS

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object.

14 TESTS WITH OIL FUEL

When oil fuel is used the rules governing the tests may be modified to conform to the characteristics of liquid fuel. The "flying" method of starting and stopping may be employed, and the duration reduced.

TABLE 1 DATA AND RESULTS OF SHOP-TEST

CODE OF 1912

(1) Test of....................locomotive at........................
  to determine.................. conducted by...................
(2) Type of locomotive......................................
(3) Type of auxiliaries.......................................
(4) Dimensions of cylinder....................................
(5) Dimensions of boiler:
  (*a*) Grate surface.......................................
  (*b*) Water-heating surface.................................
  (*c*) Superheating surface..................................
(6) Dimensions of auxiliaries.................................
(7) Diameter of exhaust nozzles...............................
(8) Date ....
(9) Duration ....hr.

Average Pressures and Temperatures

(10) Steam pressure in boiler by gage..............................lb.
(11) Steam chest pressure...lb.
(12) Temperature of feedwater....................................deg.
(13) Temperature of smoke-box....................................deg.
(14) Force of draft in smoke-box...................................in.
(15) Force of draft in furnace.....................................in.

Total Quantities

(16) Total coal as fired..........lb.
(17) Moisture in coal..........per cent
(18) Total dry coal..........lb.
(19) Ash, cinders, and refuse:
    (*a*) From ash-pan..........lb.
    (*b*) From smoke-box..........lb.
    (*c*) Total..........lb.
(20) Percentage of ash, cinders, and refuse to dry coal..........per cent
(21) Percentage of ash by analysis..........per cent
(22) Total combustible consumed (Line 18—Line 19)..........lb.
(23) Calorific value per lb. of dry coal by calorimeter test..........B.t.u.
(24) Total weight of water fed..........lb.
(25) Percentage of moisture in steam or number of degrees of superheating..........per cent or deg.
(26) Total water evaporated corrected for moisture or superheat in steam..........lb.
(27) Total equivalent evaporation from and at 212 deg..........lb.

Hourly Quantities

(28) Dry coal per hour..........lb.
(29) Dry coal per hour per sq. ft. of grate..........lb.
(30) Water evaporated per hour..........lb.
(31) Equivalent evaporation per hour from and at 212 deg..........lb.
(32) Equivalent evaporation per hour from and at 212 deg. per sq. ft. of water heating surface..........lb.

Indicator Diagrams

(33) Mean effective pressure in lb. per sq. in.:
    (*a*) First cylinder..........
    (*b*) Second cylinder..........
(34) Back pressure at mid-stroke:
    (*a*) First cylinder..........
    (*b*) Second cylinder..........

Speed

(35) Revolutions per minute..........rev.
(36) Corresponding miles per hour..........miles

Power

(37) Indicated horsepower:
    (*a*) First cylinder..........i.h.p.
    (*b*) Second cylinder..........i.h.p.
    (*c*) Whole engine..........i.h.p.
(38) Pull on drawbar..........lb
(39) Dynamometer horsepower..........dyn.h.p.
(40) Brake horsepower..........br.h.p.

Economy Results

(41) Dry coal per i.h.p. per hour..........lb.
(42) Dry coal per dynamometer h.p..........lb.

(43) Dry steam per i.h.p. per hour..............................lb.
(44) Dry steam per dynamometer h.p. hour.........................lb.

BOILER PERFORMANCE

(45) Water evaporated per lb. of dry coal...........................lb.
(46) Equivalent evaporation from and at 212 deg. per lb. of dry coal....lb.
(47) Equivalent evaporation from and at 212 deg. per lb. of combustible ....lb.
(48) Efficiency of boiler, furnace and grate.....................per cent

CODE FOR ROAD TESTS

1 OBJECT AND PREPARATIONS

The leading object of a road test as covered by this code is the determination of the coal and steam consumption of a locomotive per unit of power under the practical conditions of railroad service. The preparations for a road test are the same as those noted for the shop test, except for restrictions imposed by the regular operation of the locomotive, and in addition those required for data to be obtained on the road. See Appendix No. 34.

It is assumed that a dynamometer car is available for registering the amount of pull on the drawbar, in the manner referred to in Appendix No. 20.

To facilitate the work of the men who operate the indicators and read the instruments at the front end of the locomotive, as also to protect them from wind and rain and jolting of the locomotive, the smoke-box should be surrounded with a suitable housing or pilot-box extending back to the cylinders.

2 FUEL[1]

Select the fuel required in accordance with the special object in view. If maximum efficiency or capacity is desired, the fuel should be some kind of coal that is regarded as a standard for the locality where the test is made, as noted in the Boiler Code, page 1715.

3 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a road test of a locomotive are:

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[1] See Section 14, Page 1781, in Code for Shop Tests for memoranda relating to oil fuel.

(*a*) Platform scales for weighing coal and ashes located at the terminals, the coal being weighed preferably into sacks duly marked
(*b*) A coal calorimeter
(*c*) A water meter suitably calibrated for measuring the feedwater consumption
(*d*) A graduated scale attached to the water glass, and one or more water glasses with suitable scales attached to the water tank of the tender
(*e*) Suitable levels or plumb-lines to show the inclination of the boiler and tender
(*f*) Pressure gages, draft gages, and thermometers
(*g*) A steam calorimeter
(*h*) Steam engine indicators
(*i*) A planimeter
(*j*) A tachometer or other speed-measuring apparatus
(*k*) A dynamometer car for determining the pull on the drawbar

Directions for their use and calibration are given on pages 1691 to 1702, pages 1713 and 1714, and in Appendix No. 34.

The steam used for auxiliary purposes other than the main engine, embracing air pump, blower, calorimeter, whistle, that escaping at the safety valve, etc., may be estimated from data obtained by testing them one by one either before or after the main trial, using the water-glass method described in Appendix No. 3.

4 OPERATING CONDITIONS

In a road test the operating conditions are those pertaining to the regular service on the railroad, embracing principally the usual stops at stations, the usual speeds, and other conditions required for making schedule time, including those pertaining to the methods of firing, manipulation of the throttle valve, reverse lever, etc. If it is desired to make the test under other conditions, these should all be determined and maintained accordingly.

5 DURATION

The duration of a road test depends upon the length of the route between terminals. In fast passenger service the route should, if practicable, be at least 100 miles long. In service requiring frequent stops and in freight service the distance may be much shorter.

The length of time upon which the hourly rates of combustion and evaporation are based is the total time that the throttle valve is open, and not the elapsed time between the so-called starting and stopping times.

6 STARTING AND STOPPING

The fire having been thoroughly cleaned and the ashes and refuse removed from the ash-pan and smoke-box, burn down the fire as low as practicable just before the locomotive leaves the round house, note its average thickness, the steam pressure, the water levels, the reading of the meter, and the time, and consider the latter the starting time. Thereafter cover the fire with weighed coal and proceed with the regular work of the test. The water data should be taken just before the train starts.

As the end of the route is approached, let the fire burn down to such an extent as to leave, as nearly as practicable, after cleaning, the same amount of live coal on the grate as that observed at the start. On reaching the terminal take the pressure, the water levels, the meter reading, and the time, and consider the latter the stopping time. Then clean the fire, and if its thickness differs from that observed at the start, make a correction in the weight of coal fired accordingly. Likewise, correct for any difference in the water levels. Finally haul the ashes and refuse from the ash-pan and remove the cinders and ash from the smoke-box.

7 RECORDS

The data should be recorded in the general manner pointed out on page 1703, bearing in mind the extremely fluctuating character of the load which often obtains, and the great variations in the class of service, whether passenger or freight. Readings of the instruments concerned, so far as they can be observed when the locomotive and train are in motion, should be taken every five minutes. Indicator diagrams should be obtained one after another as fast as possible, say a set every two minutes; and at the same time the corresponding pressure, position of throttle valve and reverse lever, speed, and force of draft should be observed. The intervals of time here given may be lengthened when the service is such that the conditions are substantially uniform. Special readings of the meter, water levels and total number of sacks of coal fired should be taken at each stopping place. Careful observations should be made throughout the trial of the time of passing each mile-post, the time that the throttle valve is opened and closed, not only at each stop but also when it is necessary to shut off steam on down grades, and the time of arriving and leaving each station; also the length of time the safety valve, whistle, blower, train heating system, lighting system and other steam using apparatus are in operation. A record should be taken of the water levels before and after scooping water from track tanks the injector being stopped, and of the number of injector applications and water wasted.

8 SAMPLING AND DRYING COAL

While the coal is being weighed into the sacks, or otherwise weighed, it should be systematically sampled and its moisture determined in the manner pointed out on page 1712.

9 ASHES AND REFUSE

The ashes and refuse taken from the ash-pan and smoke-box at the close of the test should be weighed in a dry state, and a sample analyzed if desired for unburned carbon.

10 CALORIFIC TESTS OF COAL

The quality of the coal should be determined by a calorimeter test of the sample above referred to. Methods of making this test are described on page 1698 and in Appendix No. 12.

11 CALCULATION OF RESULTS

For methods of calculating the principal results, reference may be made to page 1722 of the Boiler Code and page 1735 of the Engine Code.

To determine the average drawbar pull, the dynamometer record should be averaged either by the use of a planimeter or by direct measurement at a sufficient number of points to give an equivalent result.

The maximum pull should be measured at the point of the record which shows the highest sustained pull, and not at the highest point which happens to be reached by the momentary fling of the marking pen.

12 DATA AND RESULTS

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object:

TABLE 2 DATA AND RESULTS OF ROAD-TEST

CODE OF 1912

(1) Test of....................locomotive on.........................
to determine..............conducted by.....................
(2) Type of locomotive...
(3) Type of auxiliaries..
(4) Dimensions of cylinders..
(5) Dimensions of boiler:
(*a*) Grate surface ....
(*b*) Water heating surface...................................
(*c*) Superheating surface ....
(6) Dimensions of auxiliaries......................................
(7) Diameter of exhaust nozzles....................................
(8) Weight of locomotive and tender, including water.............tons
(9) Number of cars, including dynamometer car......................
(10) Weight of cars, including dynamometer car...................tons
(11) Length of route..miles
(12) Train load expressed in ton-miles......................ton-miles
(13) Total load expressed in ton-miles......................ton-miles
(14) Schedule time between terminals.............................hr.
(15) Name of engineman ....
(16) Name of fireman ....
(17) Date ....
(18) Duration (elapsed time between starting and stopping times....hr.
(19) Actual elapsed time between terminals.......................hr.
(20) Length of time throttle valve is open.......................hr.
(21) Number of stops...
(22) Time consumed in stops......................................hr.
(23) Running time (Line 19 — Line 22)............................hr.

Average Pressures, Temperatures, etc.

(24) Steam pressure in boiler by gage............................lb.
(25) Steam chest pressure..lb.
(26) Temperature of feedwater...................................deg.
(27) Temperature of smoke-box...................................deg.
(28) Force of draft in smoke-box.................................in.
(29) Force of draft in furnace...................................in.
(30) Weather, wind, etc..
(31) Average position of throttle..................................
(32) Average position of reversing lever...........................
(33) Average number of strokes of air pump per minute..............

Total Quantities

(34) Total coal as fired...lb.
(35) Moisture in coal..per cent
(36) Total dry coal..lb.
(37) Ash, cinders, and refuse:

(*a*) From ash-pan ....lb.
(*b*) From smoke-box ....lb.
(*c*) Total ....lb.
(38) Percentage of ash, cinders, and refuse to dry coal..........per cent
(39) Percentage of ash, by analysis..........................per cent
(40) Total combustible consumed (Line 36—Line 37)...............
(41) Calorific value per lb. of dry coal by calorimeter test..........B.t.u.
(42) Total weight of water fed..............................lb.
(43) Percentage of moisture in steam or number of degrees of superheating..................................per cent. or deg.
(44) Total water evaporated corrected for moisture or superheat in steam ....lb.
(45) Estimated weight of steam used by air pump, blower, calorimeter, whistle, that escaping at the safety valve, etc...........lb.
(46) Percentage borne by steam used for auxiliary purposes named in Line 45 to total output.......................... per cent

Hourly Quantities

(47) Dry coal per hour (Line 36 ÷ Line 20).......................lb.
(48) Dry coal per hour per sq. ft. of grate.........................lb.
(49) Water evaporated per hour (Line 44 ÷ Line 20)................lb.
(50) Equivalent evaporation per hour from and at 212 deg...........lb.
(51) Equivalent evaporation per hour from and at 212 deg. per sq. ft. of water heating surface.................................lb.
(52) Dry steam consumed by engine alone per hour (Line 44 — Line 45)..lb.

Indicator Diagrams

(53) Mean effective pressure in lb. per sq. in.:
(*a*) First cylinder ....
(*b*) Second cylinder ....
(54) Back pressure at mid-stroke:
(*a*) First cylinder ....
(*b*) Second cylinder ....

Speed

(55) Revolutions per minute..................................rev.
(56) Average speed in miles per hour............................miles
(57) Maximum speed in miles per hour...........................miles

Power

(58) Indicated horsepower:
(*a*) First cylinder..................................i.h.p.
(*b*) Second cylinder..................................i.h.p.
(*c*) Whole engine..................................i.h.p.
(59) Average drawbar pull..................................lb.
(60) Maximum drawbar pull..................................lb.
(61) Dynamometer horsepower...............................dyn.h.p.

Economy Results

(62) Dry coal consumed per i.h.p. per hour.........................lb.
(63) Dry coal consumed per dynamometer h.p. per hour...............lb.

(64) Dry steam per i.h.p. per hour....................................lb.
(65) Dry steam per dynamometer per h.p.-hr..........................lb.
(66) Dry coal consumed per ton mile, train load and total load........lb.

BOILER PERFORMANCE .

(67) Water evaporated per lb. of dry coal............................lb.
(68) Equivalent evaporation from and at 212 deg. per lb. of dry coal...lb.
(69) Equivalent evaporation from and at 212 deg. per lb. of combustible ....lb.
(70) Efficiency, based on dry coal..............................per cent

PART XII

GAS PRODUCER CODE

RULES FOR CONDUCTING TESTS OF GAS PRODUCERS

1 OBJECT AND PREPARATIONS

Determine the object, take the dimensions, note the physical condition of the producer and its appurtenances, install the testing appliances, etc., as pointed out in the general instructions given on pages 1689, 1690, 1691, 1702 and 1703, and make preparations for the test accordingly.

2 FUEL[1]

Determine the character of the fuel to be used. If an untried fuel is selected and a test-producer is available, make a preliminary trial of the fuel in this apparatus and ascertain its working characteristics and the proper methods of handling it.

In tests of maximum efficiency and capacity of a producer for comparison with other producers, the fuel should be some kind of coal which is commercially regarded as a standard for such use in the locality where the test is made. The coal selected for such tests should be the best of its class and free from unusual slag-forming impurities.

The size of the coal should be determined by screening a sample, using the screens referred to on page 1692, or in Appendix No. 7.

3 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for producer tests are

(*a*) Platform scales for weighing coal and ashes
(*b*) A coal calorimeter
(*c*) A gas calorimeter
(*d*) Gas analyzing apparatus and appliances for determining tar and soot

---

[1] This code is primarily intended for producers using coal. If other fuel, such as wood or oil is burned, the rules may be modified accordingly.

(*e*) A gas meter, pitot tube, or other suitable apparatus for measuring the gas output
(*f*) A manometer or pressure gage
(*g*) Water meters for measuring feed and scrubber water, and steam meters for measuring steam used by the apparatus
(*h*) Thermometers

Full directions regarding the use and calibration of the above noted appliances are given on pages 1691 to 1702, and in various appendices there referred to.

The location of the pitot tube, if used, should be in the delivery pipe at a point near the producer or just beyond the scrubber, or at both points, according to the use made of the gas, either for fuel or power, and other requirements.

4 OPERATING CONDITIONS

Determine what the operating conditions should be to conform to the object in view, as pointed out on page 1703, and see that they prevail throughout the trial.

5 DURATION

The duration of both efficiency and capacity tests of a producer, with the exceptions noted below, should be such that the total consumption of fuel is at least ten times the weight of the fuel contained in the producer when in normal operation, estimating this weight in the case of coal at 45 lb. per cu. ft.

In cases like down-draft producers which require the fuel bed to be entirely removed and rebuilt at regular intervals, and in producers where a complete cleaning and renewal occurs before the total consumption above stipulated has been reached, the duration should be that of the regular commercial operating cycle, or the time elapsing between two successive renewals of the fuel bed.

6 STARTING AND STOPPING

The conditions regarding the temperature of the producer and its contents, and the quantity and quality of the latter, should be as nearly as possible the same at the end as at the beginning of the trial. To secure the desired equality of conditions, the starting and stopping should occur at times of regular cleanings, and they should be preceded for a period of not less than 10 hours by the same regular working conditions as those characterizing the test as a whole. The operations of starting and stopping should then be carried on as follows:

(*a*) *Up-Draft Suction-Producers.* Remove the ash and clinkers from the grate and the lower part of the furnace space, taking care that the crust or closely-united layer which supports the coal above is not unduly disturbed. Then break open the crust and allow the mass to drop into the space left vacant. Introduce a poker rod through the poke holes in the upper head and stir up the coal within, thereby causing it to settle and fill the remaining spaces. As a final step, quickly replenish the producer with coal, leaving the hopper level-full, take the time, and consider this the starting time. Then clean the ash-pit, and thereafter proceed with the regular work of the test, using weighed coal.

When the time arrives for bringing the trial to a close, the cleaning operations described above are repeated, ending with filling the hopper, taking the time, and considering this the stopping time; finally hauling the ashes and refuse from the ashpit.

(*b*) *Up-Draft Pressure Producers.* Remove the ashes until the top of the ashbed is lowered to the normal working point, say six inches above the blast-hood. Introduce the poker-rod and break down any bridge or crust that may have formed, at the same time closing up the channels that run through the fuel bed, thereby making the bed homogeneous. Then replenish the producer with coal, fill the hopper level-full, take the time, and consider this the starting time. Thereafter proceed with the regular work of the test, using weighed coal.

When the time approaches for closing the test, the operations above described are repeated, ending with replenishing the producer and filling the hopper with weighed coal, taking the time, and considering this the stopping time. The ashes and refuse finally removed are to be dried before weighing, or a sample should be taken and the moisture, as determined therefrom, allowed for.

(*c*) *Down-Draft Pressure Producer.* Thoroughly clean the producer of its entire contents. Introduce a weighed supply of coke or coal, start the fire, and build up the fuel bed to its working condition, using weighed coal. When this point is reached, take the time, and consider this the starting time. Thereafter proceed with the regular work of the test.

When the time approaches for closing the test, burn the fuel bed as low as practicable to prepare for cleaning, stop the exhauster, note the time, and consider this the stopping time. Then completely empty the producer, quench the fire remaining in the live coals, separate and weigh the coke and ash, and deduct the weight of the former from that of the coke as charged. Finally dry the ash and refuse, or take a sample and allow for the moisture determined therefrom.

7 RECORDS

8 SAMPLING AND DRYING COAL

9 ASHES AND REFUSE

10 CALORIFIC TESTS AND ANALYSES OF COAL

The directions pertaining to the above divisions of the subject are practically the same as those given under the corresponding headings in the Boiler Code, and reference may be made to pages 1720 and 1721 of that code for these directions.

11 CALORIFIC TESTS AND ANALYSES OF GAS OUTPUT

The quality of the gas should be determined by calorific tests and analyses, continuous samples for this purpose being taken from the delivery pipe at a point near the producer and at other points as may be needed.

The calorific test should be made with the Junker calorimeter, described in Appendix No. 13, or its equivalent. Unless otherwise required the "higher value" should be employed in calculating the results of the test. For an approximate determination of the composition of the gas, the Orsat apparatus may be used, and for complete determinations, the Hempel apparatus or its equivalent. Both of the two named are described in Appendix No. 15. The frequency with which these determinations should be made depends on the uniformity of the output, but the intervals, where practicable, should not be more than one-half hour, the time taken for collecting each sample being not less than one-half hour.

12 CALCULATION OF RESULTS

(*a*) *Total Volume of Gas Delivered.* The volume of gas found by pitot tube measurement is determined by multiplying the area of the delivery pipe in sq. ft. at the tube by the velocity of the gas in ft. per minute, and the product by the duration of the trial in minutes. The equivalent volume at atmospheric pressure (30 in. barometer) and temperature of 62 deg. fahr., is obtained by multiplying the absolute pressure of the gas in lb. per sq. in. (gage pressure plus 14.7) by the constant 35.3, and dividing the product by the absolute temperature of the gas (temperature by thermometer plus 460 deg.).

(*b*) *Net Volume of Gas Delivered.* The net volume of gas delivered is found by subtracting from the total volume the equivalent volume of gas required for furnishing steam drawn from an outside source, if any, or for furnishing power used for any purpose concerned in the operation of the producer and its auxiliaries.

(c) *Weight of Gas.* The weight of dry gas delivered is found by multiplying the volume in cu. ft., reduced to 62 deg. and 30 in., given in Table 2, Line 21, by the weight per cu. ft. of gas given in Table 2, Line 77.

The weight of the gas per cu. ft. is determined by multiplying the percentage of each component gas as found by analysis (see Lines 55 to 63, Table 2) by its weight in lb. per cu. ft., at 62 deg. and 30 in., as given in the following table, and dividing the sum of the products by 100.

CO_2	0.1116	CH_4	0.0428
CO	0.0736	C_2H_4	0.0737
O	0.0842	SO_2	0.1638
H	0.0053	H_2S	0.0868
N	0.0740		

(d) *Moisture in Gas Leaving Producer.* The percentage of moisture in the gas is found by passing a measured sample of the gas through a chloride of calcium tube and weighing the amount of moisture absorbed.

(e) *Percentage of Tar and Soot in Gas.* The percentage of tar and soot is found by comparing the total weight determined, including that collected from the various tar drips with the total weight of dry fuel used.

(f) *Efficiency.* The efficiency is the relation between the calorific value of the gas per lb. of fuel charged, or combustible burned, and the calorific value of 1 lb. of fuel or combustible. The former is ascertained by multiplying the B.t.u. per cu. ft. of gas as determined by the calorimeter test (higher value) by the cu. ft. of gas delivered, and dividing the product by the total weight of fuel charged or combustible burned.

The "combustible burned" is determined by subtracting from the weight of coal charged the moisture in the coal and the ash and refuse, including unburned coal which is withdrawn from the producer or ash-pit during the progress of the trial. The "combustible" used for determining the calorific value is the weight of the coal less the moisture and ash found by analysis.

The efficiency of "conversion and cleaning" in the above calculation is found by using the total volume of gas delivered. The "efficiency of the plant" is found by using the net volume of gas delivered.

(g) *Heat Balance.* The various quantities showing the distribution of heat in the heat balance given in Table 2, are computed in the following manner:

The heat contained in the dry gas is found by multiplying the cubic feet of gas at 62 deg. and 30 in. per lb. of dry coal by the calorific value of 1 cu. ft. of gas at 62 deg. and 30 in. (higher value).

The heat carried away by the scrubber is obtained by multiplying the weight of water fed to the scrubber by the number of degrees rise of temperature, and dividing the product by the total weight of dry coal consumed.

The heat contained in the moisture leaving the producer is found by multiplying the total weight of dry gas by the proportion of moisture in the gas leaving the producer and by the total heat of 1 lb. of superheated steam at the temperature of the gas leaving the producer reckoned from the temperature of the air in the room, and dividing the product by the weight of dry coal consumed.

13 DATA AND RESULTS

The data and results should be reported in accordance with either the Short Form or Complete Form given herewith, adding lines for data not provided for or omitting those not required as may conform to the object in view.

If a preliminary trial of the fuel is made in a test-producer, add to the table the general results obtained.

14 CHART

In trials having for an object the determination and exposition of the complete performance from beginning to end, the entire log of readings and data should be plotted on a chart and represented graphically.

TABLE 1 DATA AND RESULTS OF GAS PRODUCER TEST

SHORT FORM, CODE OF 1912

(1) Test of..................producer located at....................
to determine....................conducted by...............
(2) Type of producer..
(3) Rated capacity of producer..................................
(4) Date ....
(5) Duration ....hr.
(6) Kind of coal[1] and where mined..............................
(7) Size of coal...

Average Pressures, Temperatures, Etc.

(8) Steam pressure in vaporizer by gage........................lb.
(9) Gas pressure in delivery main at point where gas is measured...................................in. water
(10) Temperature of feedwater..................................deg.

---

[1] If other fuel than coal is used, the lines may be changed to read accordingly.

(11) Temperature of gas in delivery main near producer............deg.
(12) Temperature of gas in delivery main at point where gas is measured ....deg.
(13) Force of blast or draft in ashpit...............................in.

Total Quantities.

(14) Weight of coal as charged.....................................lb.
(15) Percentage of moisture in coal.........................per cent
(16) Total weight of dry coal consumed.............................lb.
(17) Total ash and refuse..lb.
(18) Percentage of ash and refuse in dry coal:..................per cent
(19) Total cu. ft. of gas as measured...........................cu. ft.
(20) Equivalent cu. ft. of gas at 62 deg. and 30 in................cu. ft.
(21) Net cu. ft. of gas at 62 deg. and 30 in.[1].....................cu. ft.
(22) Total water fed to vaporizer..................................lb.
(23) Total water supplied to scrubber..............................lb.

Hourly Quantities

(24) Dry coal consumed per hour....................................lb.
(25) Dry coal per sq. ft. of main fuel bed per hour.................lb.
(26) Total cu. ft. of gas delivered per hour.......................cu. ft.
(27) Total cu. ft. of gas per hour at 62 deg. and 30 in............cu. ft.
(28) Net cu. ft. of gas per hour at 62 deg. and 30 in..............cu. ft.

Economy Results

(29) Total cu. ft. of gas delivered per lb. of dry coal (Line 13 ÷ Line 10)..cu. ft.
(30) Equivalent total gas at 62 deg. and 30 in. per lb. of dry coal...cu. ft.
(31) Net cu. ft. of gas at 62 deg. and 30 in. per lb. of dry coal[1]......cu. ft.
(32) Net cu. ft. of gas at 62 deg. and 30 in. per lb. of combustible...cu. ft.

Efficiency[2]

(33) Calorific value of dry coal per lb..........................B.t.u.
(34) Calorific value of combustible per lb.......................B.t.u.
(35) Calorific value of gas per cu. ft. (higher value).............B.t.u.
(36) Efficiency of producer based on coal.......................per cent
(37) Efficiency of producer based on combustible................per cent

Cost of Production

(38) Cost of coal per ton of .... lb. delivered.................dollars
(39) Cost of coal required for producing 10,000 net cu. ft. of gas at 62 deg. and 30 in...dollars
(40) Cost of coal required for producing one million B.t.u. in the gas ....dollars

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[1] After deducting equivalent gas required for auxiliaries.

[2] If the efficiency is based on the "low value" of the heat units in the gas the fact should be so stated.

TABLE 2 DATA AND RESULTS OF GAS PRODUCER TEST

COMPLETE FORM, CODE OF 1912

(1) Test of..................producer located at....................
to determine ....conducted by................

Dimensions

(2) Outside diameter of producer....................................ft.
(3) Height of producer...ft.
(4) Inside diameter of producer.....................................ft.
(5) Diameter of grate...ft.
(6) Area of grate..sq. ft.
(7) Percentage of air space in grate..........................per cent
(8) Area of fuel bed (at maximum diameter)...................sq. ft.
(9) Area of water heating surface in vaporizer..................sq. ft.
(10) Rated capacity of producer in lb. of coal per hour...............lb.
(11) Date ....
(12) Duration ....hr.
(13) Kind of coal[1] and where mined...................................
(14) Size of coal...

Average Pressures, Temperatures, Etc.

(15) Steam pressure in vaporizer by gage...........................lb.
(16) Gas pressure in main at point where gas is measured......in. water
(17) Force of blast or draft in ashpit.........................in. water
(18) Barometric pressure.......................................in. merc.
(19) Temperature of feedwater entering vaporizer..................deg.
(20) Temperature of gas in main near producer....................deg.
(21) Temperature of gas in main at point where gas is measured.....deg.
(22) Temperature of air in room...deg.
(23) Temperature of water entering scrubber.......................deg.
(24) Temperature of water leaving scrubber........................deg.
(25) Weight of dry gas per cu. ft. reduced to 62 deg. and 30 in........lb.

Total Quantities

(26) Weight of coal as fired..lb.
(27) Percentage of moisture in coal.................................per cent
(28) Total weight of dry coal consumed................................lb.
(29) Total ash and refuse...lb.
(30) Percentage of ash and refuse in dry coal.....................per cent
(31) Total number of cu. ft. of gas as measured.....................cu. ft.
(32) Equivalent cu. ft. of gas at temperature of 62 deg. and pressure of atmosphere of 30 in...cu. ft.
(33) Net cu. ft. of gas at 62 deg. and 30 in.[2]........................lb.
(34) Total weight of dry gas...lb.
(35) Percentage of moisture in gas leaving producer.........per cent

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[1] If other fuel than coal is used the lines may be changed to read accordingly.

[2] After deducting equivalent gas required for auxiliaries.

(36) Percentage of tar and soot in gas referred to total fuel....per cent
(37) Total water fed to vaporizer....................................lb.
(38) Total water evaporated in vaporizer.............................lb.
(39) Total weight of steam supplied to producer......................lb.
(40) Total weight of water fed to scrubber..........................

Hourly Quantities

(41) Dry coal consumed per hour.......................................lb.
(42) Dry coal consumed per hour per sq. ft. of grate.................lb.
(43) Dry coal consumed per hour per sq. ft. of main fuel bed.......lb.
(44) Total cu. ft. of gas delivered per hour (Line 20 ÷ Line 17)...cu. ft.
(45) Total cu. ft. of gas per hour at 62 deg. and 30 in..........cu. ft.
(46) Net cu. ft. of gas delivered per hour at 62 deg. and 30 in......cu. ft.
(47) Weight of dry gas per hour.......................................lb.
(48) Water fed per hour to vaporizer..................................lb.
(49) Water evaporated per hour in vaporizer...........................lb.
(50) Steam supplied to producer per hour..............................lb.
(51) Water fed to scrubber per hour...................................lb.

Proximate Analysis of Coal

(52) Fixed carbon...per cent
(53) Volatile matter..per cent
(54) Moisture ....per cent
(55) Ash ....per cent

100 per cent

(56) Sulphur, separately determined.........................per cent

Ultimate Analysis of Dry Coal

(57) Carbon (C)...per cent
(58) Hydrogen (H)...per cent
(59) Oxygen (O)...per cent
(60) Nitrogen (N)...per cent
(61) Sulphur (S)..per cent
(62) Ash ....per cent

100 per cent

(63) Moisture in sample of coal as received.................per cent

Analysis of Ash and Refuse

(64) Carbon ....per cent
(65) Earthy matter..per cent

Analysis of Gas by Volume

(66) Carbon dioxide (CO_2)................................per cent
(67) Carbon monoxide (CO)...................................per cent
(68) Oxygen (O)...per cent
(69) Hydrogen (H)...per cent
(70) Marsh gas (CH_2).....................................per cent
(71) Olefiant gas (C_2H_4)................................per cent
(72) Sulphur dioxide (SO_2)...............................per cent
(73) Hydrogen sulphide (H_2S).............................per cent
(74) Nitrogen (N by difference).............................per cent

100 per cent

Calorific Values by Calorimeter

(75) Calorific value of dry coal per lb..............................B.t.u.

(76) Calorific value of combustible per lb..........................B.t.u.

(77) Calorific value of gas per cu. ft. at 62 deg. and 30 in. (higher value) ....B.t.u.

Economy Results

(78) Total cu. ft. of gas as measured, per pound of dry coal consumed ....cu. ft.

(79) Equivalent cu. ft. of gas at 62 deg. and 30 in. per lb. of dry coal.cu. ft.

(80) Equivalent cu. ft. of gas at 62 deg. and 30 in. per lb. of combustible ....cu. ft.

(81) Net cu. ft. of gas at 62 deg. and 30 in. per lb. of dry coal.....cu. ft.

(82) Net cu. ft. of gas at 62 deg. and 30 in. per lb. of combustible..cu. ft.

Efficiency[1]

(83) Efficiency of producer based on coal:
- (*a*) Conversion and cleaning..........................per cent
- (*b*) Plant ....per cent

(84) Efficiency of producer based on combustible:
- (*a*) Conversion and cleaning..........................per cent
- (*b*) Plant ....per cent

Cost of Production

(85) Cost of coal per ton of .... lb., delivered....................dollars

(86) Cost of coal required for producing 10,000 net cu. ft. of gas at 62 deg. and 30 in..dollars

(87) Cost of coal for producing 1,000,000 B.t.u....................dollars

Heat Balance Based on 1 Lb. of Dry Coal

		B.t.u.	Per cent
(88)	Heat contained in dry gas......................	......	......
(89)	Heat carried away by scrubber..................	......	......
(90)	Heat contained in moisture leaving producer......	......	......
(91)	Heat unaccounted for, including radiation,—difference between the sum of Lines 88, 89, and 90 and Line 92..............................	......	......
(92)	Total calorific value of 1 lb. of dry coal, same as Line 75..................................	......	......

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[1] If the efficiency is based on the "low value" of the heat units in the gas, the fact should be so stated.

PART XIII

GAS AND OIL ENGINE CODE

RULES FOR CONDUCTING TESTS OF GAS AND OIL ENGINES

1 OBJECT AND PREPARATIONS

Determine the object, take the dimensions, note the physical condition of the engine and its appurtenances, install the testing appliances, etc., as explained in the general instructions given on pages 1689, 1690, 1691, 1702 and 1703, and make preparations for the test accordingly.

2 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for simple performance tests of gas and oil engines are:

(*a*) Tanks and platform scales for weighing oil
(*b*) A calorimeter for determining the heat of combustion of oil
(*c*) A gas meter or other apparatus for measuring gas
(*d*) A gas calorimeter
(*e*) Pressure gages and thermometers
(*f*) Gas engine indicators
(*g*) A planimeter
(*h*) A tachometer or other speed-measuring apparatus

and in addition for the determination of the complete performance

(*i*) Gas analyzing apparatus
(*j*) A water meter for measuring jacket water
(*k*) A friction brake or dynamometer

Full directions regarding the use and calibration of these appliances are given on pages 1691 to 1702, and in various appendices there referred to.

3 OPERATING CONDITIONS

Determine what the operating conditions should be to conform to the object in view, and see that they prevail throughout the trial, as pointed out on page 1703, Part I.

4 DURATION

The test of a gas or oil engine with substantially constant load should be continued for such time as may be necessary to obtain a number of successive records covering periods of half an hour or less during which the results are found to be uniform. In such cases a duration of three to five hours is sufficient for all practical purposes.

5 STARTING AND STOPPING

The engine having been set to work under the prescribed conditions and thoroughly heated (except in cases where the object is to obtain the performance under working conditions), the test is begun at a certain predetermined time by commencing to weigh the oil, or measure the gas, as the case may be, and take other data concerned; after which the regular measurements and observations are carried forward until the end. When the stopping time arrives the test is closed by simply taking the final readings.

6 RECORDS

The general data should be taken and recorded in the same manner as that described in the Steam Engine Code, on page 1735, to which reference may be made.

7 CALORIFIC TESTS AND ANALYSES

The quality of the oil or gas should be determined by calorific tests and analyses made on representative samples.

Directions for these tests or analyses are given on pages 1697 and 1698 under the headings Coal Calorimeters, Gas Calorimeters, Coal Analysis Apparatus, and Gas Analysis Apparatus, and in Appendices Nos. 12, 13, 14, and 15 there referred to.

8 CALCULATION OF RESULTS

(*a*) *Heat Consumption.* The number of heat units consumed by the engine is found by multiplying the heat units per lb. of oil or per cu. ft. of gas (higher value), as determined by calorimeter test, by the total weight of oil in lb. or volume of gas in cu. ft. consumed during the trial.

(*b*) *Horsepower and Efficiency.* The indicated horsepower, brake horsepower, and efficiency are computed by the same methods as those explained in the Steam Engine Code, on page 1736, to which reference may be made.

(*c*) *Heat Balance.* The various quantities showing the distribution of heat in the heat balance given in Table 2 are computed in the following manner:

The heat converted into work per i.h.p.-hr. (2545 B.t.u.) is found by dividing the work representing 1 h.p., or 1,980,000 ft-lb., per hour by the number of ft.-lb. representing 1 B.t.u., or 778.

The heat rejected in the cooling water is obtained by multiplying the weight of water supplied by the number of degrees rise of temperature, and dividing the product by the indicated horsepower.

The heat rejected in the dry exhaust gases per i.h.p.-hr. is found by multiplying the weight of these gases per i.h.p-hr. by the sensible heat of the gas reckoned from the temperature of the air in the room and by its specific heat. The weight of the dry gases per i.h.p.-hr. may be found by multiplying the weight of dry gas per lb. of carbon, using the formula:

$$\frac{11\,CO_2 + 8\,O + 7\,(CO + N)}{3\,(CO_2 + CO)}$$

in which CO_2, CO, O, and N are expressed in percentages by volume, by the proportion borne by the carbon in the total fuel, (either gas or oil), and by the weight of fuel per i.h.p-hr.

The heat lost in the moisture formed by the burning of hydrogen in the gas is found by multiplying the total heat of 1 lb. of superheated steam at the temperature of the gas, reckoning from the temperature of the air in the room, by the proportion of the hydrogen in the fuel as determined from the analysis, and multiplying the result by 9.

The heat lost through incomplete combustion is obtained by analyzing the exhaust gases and computing the heat of the unburned products which would have been produced by their combustion.

9 DATA AND RESULTS

The data and results should be reported in accordance with either the Short Form or Complete Form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view[1]:

TABLE 1 DATA AND RESULTS OF GAS OR OIL ENGINE TEST

SHORT FORM, CODE OF 1912

(1) Test of....................engine, located at....................
to determine............., conducted by....................
(2) Type and class of engine and number of cycles....................
(3) Dimensions:
  (*a*) Single or double acting..............................
  (*b*) Diameter of cylinders..............................in.
  (*c*) Stroke of pistons..............................ft.
  (*d*) Diameter of piston rods..............................in.
  (*e*) Compression space or clearance..................per cent
  (*f*) H.p. constant for 1 lb. m.e.p. and 1 r.p.m.................

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[1] See note page 1744 for reference to engines driving electric generators and other machinery.

(4) Rated capacity ....

(5) Date ....

(6) Duration ....hr.

(7) Kind of gas..

(8) Kind of oil..

(9) Physical properties of oil (specific gravity, burning point, and flashing point)..

Average Pressures and Temperatures

(10) Pressure of gas near meter..in.

(11) Barometric pressure..lb.

(12) Temperature of cooling water:

  (*a*) Inlet ....deg.

  (*b*) Outlet ....deg.

(13) Temperature of gas near meter..deg.

(14) Temperature of air..deg.

(15) Temperature of exhaust gases..deg.

Total Quantities

(16) Gas or oil consumed..cu. ft.-lb.

(17) Cooling water supplied to jackets..lb.

(18) Calorific value of gas per cu. ft., or of oil per lb., by calorimeter test (higher value)..b.t.u.

Hourly Quantities

(19) Gas or oil consumed per hour..cu. ft.-lb.

(20) Cooling water supplied to jackets per hour..lb.

Indicator Diagrams

(21) Maximum pressure ....lb.

(22) Mean effective pressure..lb.

Speed and Explosions

(23) Revolutions per minute..rev.

(24) Average number of explosions per minute..

Power

(25) Indicated horsepower..i.h.p.

(26) Brake horsepower..br. h.p.

(27) Friction horsepower by difference Line 25 — Line 26..........fr. h.p.

(28) Friction horsepower by friction diagrams..fr. h.p.

(29) Percentage of indicated horsepower lost in friction (Line 27), per cent

Economy Results

(30) Heat units consumed by engine per hour[1]:

  (*a*) Per indicated horsepower ....B.t.u.

  (*b*) Per brake horsepower ....B.t.u.

(31) Pounds of oil or cubic feet of gas consumed per hour:

  (*a*) Per indicated horsepower ....lb-cu. ft.

  (*b*) Per brake horsepower ....lb-cu. ft.

Sample Diagrams

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[1] If these results in the case of a gas engine are based on the "low value" of the heat units, that fact should be so stated.

NOTE: For an engine driving an electric generator the form may be enlarged to include electrical data in the manner given in the Steam Turbine Code, page 1749.

TABLE 2 DATA AND RESULTS OF GAS OR OIL ENGINE TEST

COMPLETE FORM, CODE OF 1912

(1) Test of....................engine, located at......................
to determine................., conducted by...................
(2) Type of engine, whether oil or gas..................................
(3) Class of engine (mill, marine, motor for vehicle, pumping, or other) ....
(4) Number of revolutions for one cycle, and class of cycle.............
(5) Method of ignition..
(6) Name of builders...
(7) Dimensions:
(*a*) Class of cylinder, whether working or compressing........
(*b*) Vertical or horizontal...............................
(*c*) Single or double acting..............................
(*d*) Diameter of cylinders...............................in.
(*e*) Stroke of pistons...................................ft.
(*f*) Diameter of piston rods.............................in.
(*g*) Compression space or clearance........................
(*h*) H.p. constant for 1 lb. m.e.p. and 1 r.p.m................
(8) Rated capacity ....
(9) Date ....
(10) Duration ....hr.
(11) Kind of oil...
(12) Physical properties of oil (specific gravity, burning point, flashing point)...
(13) Kind of gas..

Average Pressures and Temperatures

(14) Pressure of gas near meter..............................in.
(15) Barometric pressure.....................................in.
(16) Temperature of cooling water:
(*a*) Inlet ....deg.
(*b*) Outlet ....deg.
(17) Temperature of gas near meter...........................deg.
(18) Temperature of air:
(*a*) By dry-bulb thermometer...........................deg.
(*b*) By wet-bulb thermometer...........................deg.
(19) Temperature of exhaust gases............................deg.

Total Quantities

(20) Gas or oil consumed..............................cu. ft.-lb.
(21) Air supplied in cu. ft..................................cu. ft.
(22) Cooling water supplied to jackets..........................lb.
(23) Calorific value of oil per lb., or of gas per cu. ft., by calorimeter test (higher value)...............................B.t.u.

Hourly Quantities

(24) Gas or oil consumed per hour.......................cu. ft-lb.
(25) Cooling water supplied per hour............................lb.

ANALYSIS OF OIL

(26) Carbon (C) .... per cent
(27) Hydrogen (H) .... per cent
(28) Oxygen (O) .... per cent
(29) Sulphur (S) .... per cent
(30) Moisture .... per cent

100 per cent

ANALYSIS OF GAS BY VOLUME

(31) Carbon dioxide (CO_2) .... per cent
(32) Carbon monoxide (CO) .... per cent
(33) Oxygen (O) .... per cent
(34) Hydrogen (H) .... per cent
(35) Marsh gas (CH_4) .... per cent
(36) Olefiant gas (C_2H_4) .... per cent
(37) Sulphur dioxide (SO_2) .... per cent
(38) Hydrogen sulphide (H_2S) .... per cent
(39) Nitrogen (N by difference) .... per cent

100 per cent

INDICATOR DIAGRAMS

(40) Pressure in lb. per sq. in. above atmosphere:
 (*a*) Maximum pressure .... lb.
 (*b*) Pressure at beginning of stroke .... lb.
 (*c*) Pressure at end of expansion .... lb.
 (*d*) Exhaust pressure at lowest point .... lb.
(41) Mean effective pressure in lb. per sq. in .... lb.

SPEED AND EXPLOSIONS

(42) Revolutions per minute .... rev.
(43) Average number of explosions per minute ....
(44) Variation of speed between no load and full load .... rev.
(45) Fluctuation of speed on suddenly changing from full load to no load measured by the increase in the revolutions due to the change ....

POWER

(46) Indicated horsepower .... i.h.p.
(47) Brake horsepower .... br.h.p.
(48) Friction horsepower by difference (Line 46 — Line 47) ..... fr. h.p.
(49) Friction horsepower by friction diagrams .... fr. h.p.
(50) Percentage of indicated horsepower lost in friction (Line 48), per cent

Economy Results

(51) Heat units consumed by engine per hour[1]:
- (*a*) Per indicated horsepower..........................B.t.u.
- (*b*) Per brake horsepower..........................B.t.u.

(52) Pounds of oil or cubic feet of gas consumed per hour:
- (*a*) Per indicated horsepower..........................lb-cu. ft.
- (*b*) Per brake horsepower ....lb-cu. ft.

Efficiency

(53) Thermal efficiency ratio:
- (*a*) Per indicated horsepower..........................per cent
- (*b*) Per brake horsepower ....per cent

Work Done Per Heat Unit

(54) Ft-lb. of net work per b.t.u. consumed (1,980,000 ÷ Line 51*b*)...ft-lb.

Heat Balance Based on B.t.u. per i.h.p. per Hour

		B.t.u.	Per Cent
(55)	Heat converted into work......................	2545	......
(56)	Heat rejected in cooling water..................	......	......
(57)	Heat rejected in the exhaust gases..............	......	......
(58)	Heat lost due to moisture formed by burning of hydrogen	......	......
(59)	Heat lost by incomplete combustion.............	......	......
(60)	Heat unaccounted for, including radiation.......	......	......
(61)	Total heat consumed per i.h.p. per hour, same as Line 51*a*.................................	......	......

Sample Diagrams

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[1] If these results, in the case of a gas engine, are based on the low value of the heat units, that fact should be so stated.

Note: For an engine driving an electric generator, the form may be enlarged to include electrical data in the manner given in the Steam Turbine Code, page 1749.

PART XIV

CODE FOR GAS POWER PLANTS

RULES FOR CONDUCTING TESTS OF COMPLETE GAS POWER PLANTS

1 OBJECT AND PREPARATIONS

The usual object of testing a complete gas power plant, embracing producer, engine, and appurtenances, is the determination of its commercial performance, i. e., the number of lb. of fuel consumed per unit of work done in a unit of time, and the rules given in this code apply to tests having that object. For directions pertaining to tests of the producer and engine individually, reference may be made to the Producer and Gas Engine Codes, pages 1791 and 1803.

Read the general instructions given on pages 1689, 1690, 1691, 1702 and 1703, take the dimensions, note the physical conditions of the plant, install the testing appliances, etc., as there explained, and prepare for the test accordingly.

2 FUEL

Determine the character of the fuel to be used as pointed out in the Gas Producer Code, page 1791.

3 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a simple performance test of a gas power plant are:

(*a*) Platform scales for weighing coal and ashes
(*b*) A coal calorimeter
(*c*) Gas engine indicators
(*d*) A tachometer or other speed-measuring apparatus
(*e*) Wattmeters, in the case of an electric plant

Directions regarding the use and calibration of these appliances are given on pages 1691 to 1702.

If the test involves the determination of the individual performance of the producer and engine, additional instruments are required as pointed out in the Producer and Gas Engine Codes.

4 OPERATING CONDITIONS

Determine what the operating conditions should be as explained on page 1703 and see that they prevail throughout the trial.

5 DURATION

The duration of a gas producer plant test should conform to that of the producer alone, rules pertaining to which may be found in the gas producer code.

In cases where the engine is in operation only a part of the day, the hourly consumption of coal from which the economy results are computed should be the total coal burned in the producer divided by the number of hours that the engine is in operation at its working speed.

6 STARTING AND STOPPING

The rules for starting and stopping a complete plant test are governed by those required for starting and stopping the test of the producer, which are those given in the Producer Code, page 1791.

7 RECORDS

8 SAMPLING AND DRYING COAL

9 ASHES AND REFUSE

10 CALORIFIC TESTS AND ANALYSES OF COAL

11 CALCULATION OF RESULTS

The directions pertaining to the above five divisions of the subject are practically the same as those given under the corresponding headings in the Code for Complete Steam Power Plants, pages 1753 and 1754.

12 DATA AND RESULTS

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view.

DATA AND RESULTS OF TEST OF COMPLETE GAS POWER PLANT

CODE OF 1912

(1) Test of....................gas power plant at...................
to determine..............., conducted by....................
(2) Type and dimensions of producers..............................
(3) Rated capacity of producers...................................
(4) Total area of main fuel bed at maximum diameter...........sq. ft.
(5) Type and dimensions of engine.................................
(6) Rated power of engine...
(7) Date ....
(8) Duration ....hr.
(9) Kind of coal..
(10) Size of coal..

Average Pressures and Temperatures

(11) Pressure of gas near throttle valve..........................in.
(12) Barometric pressure...in.
(13) Temperature of cooling water leaving engine..................deg.
(14) Temperature of air in room...................................deg.

Total Quantities

(15) Weight of coal as charged....................................lb.
(16) Percentage of moisture in coal..........................per cent
(17) Total weight of dry coal consumed.............................lb.
(18) Total ash and refuse..lb.
(19) Percentage of ash and refuse in dry coal..................per cent
(20) Calorific value of 1 lb. of dry coal by calorimeter test......B.t.u.
(21) Cost of coal per ton of .... lb..........................dollars

Hourly Quantities

(22) Dry coal consumed per hour....................................lb.
(23) Dry coal per sq. ft. of main fuel bed per hour...................lb.

Indicator Diagrams

(24) Mean effective pressure in lb. per sq. in......................lb.

Speed and Explosions

(25) Revolutions per minute..rev.
(26) Number of explosions per minute...............................

Power

(27) Indicated horsepower developed by engine...................i.h.p.

Economy Results

(28) Dry coal consumed per i.h.p. per hour........................lb.
(29) Cost of coal per i.h.p. per hour.........................dollars
(30) Heat units consumed per i.h.p. per hour (Line 20 × Line 28)....B.t.u.

Note: For an engine driving an electric generator, the form may be enlarged to include electrical data in the manner given in the code for Complete Steam Power Plants, page 1755.

PART XV

WATERWHEEL CODE

RULES FOR CONDUCTING TESTS OF WATERWHEELS

1 INTRODUCTION

Waterwheel tests may be divided into two classes, one of which may be termed "shop" tests and the other "field" tests. The former refer to those which are conducted in a plant devoted exclusively to testing work, and the latter to tests of the wheel in its permanent location. The Holyoke Water Power Company's testing flume is an example of a shop-testing plant, being one which is equipped for turbine water wheels of any size up to 300 h.p. at 18-ft. head. This plant, it is understood, is also the only one of the kind in the country which is available for commercial work. Under these circumstances there seems to be no call at the present time (1912) for a general code of rules applying to tests of that character. The tests to which the following code refers are therefore limited to field tests; the wheel being in place, and operating so far as possible under the conditions of service for which it was installed.

2 OBJECT AND PREPARATIONS

The usual object of a waterwheel test in the field is the determination of the capacity and efficiency of the wheel at various gate openings and if practicable at various speeds, as compared with standard or guaranteed performance. Having determined the object, whatever it may be, take the dimensions, note the physical condition of the wheel and of the plant throughout, install the testing appliances, etc., following the general instructions given on pages 1689, 1690, 1691, 1702 and 1703, so far as they pertain to the work in hand, and make preparations for the test accordingly.

The most important preparations are those which relate to the determination of the power developed by the wheel, and the quantity of water which it consumes. The nature of these preparations is governed altogether by the character of the equipment. As regards power determination, the simplest method is the one applying to a case where the wheel drives an electric generator and the power is measured by calculation from the electrical output. Another simple method is one which may be used where the wheel serves as an auxiliary to steam power, and the output is determined by ascertaining the difference between the indicated horsepower developed by the engine when the wheel is on and that developed when the wheel is off. Another method which is applicable to almost any situation where there is room, although the most difficult of the three, is the use of a friction brake attached to the waterwheel shaft, being arranged so as to take the place of a section of the shaft which may temporarily be removed. As to preparation for water measurement, the desirability of preserving the maximum head of water usually makes it necessary to gage the stream supplied to the wheel or the stream leaving it, and to select or prepare for this purpose a sufficient length of canal having a uniform cross-section to determine the required velocity by float measurement. Another method consists in the use of current meters or pitometers which have been properly calibrated. In cases where some part of the head may be sacrificed either in the head race or tail race, the measurement may be made by the insertion of a suitable weir.

3 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a capacity and efficiency test of a waterwheel are:

(*a*) A friction brake, steam engine indicators, or electrical instruments, depending on the character of the equipment
(*b*) Graduated scales showing the heights of water in the flume above the wheel and in the discharge pit beneath
(*c*) One or more current meters or other apparatus for ascertaining the velocity of the water; or a weir

Directions for the use of these appliances may be found on pages 1699 to 1701 and in Appendices Nos. 18, 19, 21, and 5.

It is of the greatest importance that the water measured is that which is consumed wholly by the wheel. If water leaks by without going through the wheel, the quantity of leakage should be determined by independent measurement when the wheel is entirely shut off; in which case the gross quantity is corrected accordingly.

4 DURATION

The duration of a simple efficiency test of a waterwheel depends mainly upon the method of water measurement employed, and the time required to obtain a sufficient number of observations to insure a reliable average. After the desired load and other conditions have been obtained, continuous observations and measurements for a period of 15 minutes is sufficient for all practical purposes, provided the water is measured by a weir, but a longer time is necessary when other methods of measurement are used.

5 RECORDS

The records should be obtained in a manner conforming to the principles explained on page 1703. Readings of the weight on the brake arm, levels of water in the flume and discharge pit, indications of the current meters, and revolutions per minute, should be taken every five minutes, and at more frequent intervals if they show much fluctuation. In case of float measurement, repeated observations should be made one after the other throughout the whole period of the trial.

6 CALCULATION OF RESULTS

The total average head of water on the wheel is obtained by adding together the reading of the scale in the flume and the vertical distance between the zero of this scale and that of the scale in the discharge pit, and subtracting the reading of the latter scale, both readings being taken in reasonably still water.

The velocity of water in the measuring canal is found by averaging the readings obtained at several points extending over the whole width of the canal.

The cubic feet of water consumed per minute is obtained by multiplying the cross-section of the stream in square feet by the velocity of the water in feet per minute, determined as stated above.

The total power of water available is obtained by multiplying the net weight of water in pounds consumed per minute by the total average head in feet on the wheel, and dividing the product by 33,000.

The brake horsepower delivered by the wheel is found by multiplying the net weight on the brake arm in pounds by the circumference of the corresponding circle in feet and by the number of revolutions per minute, and dividing the final product by 33,000.

7 DATA AND RESULTS

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for or omitting those not required, as may conform with the object in view:

DATA AND RESULTS OF WATERWHEEL TEST ADAPTED TO BRAKE MEASUREMENT OF POWER

CODE OF 1912

(1) Test of..................water wheel located at..................
to determine................., conducted by..................

(2) Type of wheel and class of service..............................

(3) Type of generator, if any, kind of current, etc....................

(4) Rated power of wheel.......................................h.p.

(5) Cross-section of stream where velocity of water is measured...sq. ft.

General Data

(6) Date ....

(7) Duration of period covered by test..........................hr.

(8) Average net weight on brake arm.............................lb.

(9) Average revolutions per minute..........................r.p.m.

(10) Total average head of water on wheel.........................ft.

(11) Average velocity of water per minute in measuring canal........ft.

(12) Cu. ft. of water consumed per minute.....................cu. ft.

(13) Weight of water consumed per minute (Line 12 × 62.35)..........lb.

(14) Leakage per minute..lb.

(15) Net water consumed per minute (Line 13 — Line 15)...........lb.

Power

(16) Total power of water available.............................h.p.

(17) Brake horsepower delivered by wheel.....................br. h.p.

Efficiency

(18) Efficiency of wheel (Line 17 ÷ Line 16)..................per cent

THIRD SECTION

PART XVI.

APPENDICES

APPENDICES

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Appendix No. 2. Leakage Tests of Engines.
Appendix No. 3. General Precautions Regarding Leakage.
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APPENDICES

APPENDIX NO. 1

CLEARANCE MEASUREMENT BY WATER

To measure the clearance by actual test, the engine is carefully set on the center, with the piston at the end where the measurement is to be taken. Assuming, for example, a Corliss engine, the best method is to remove the steam valve so as to have access to the whole steam port, and then fill up the clearance space with water, which is poured into the open port through a funnel. The water is drawn from a receptacle containing a quantity previously measured. When the whole space, including the port, is completely filled, the quantity left is measured, and the difference shows the amount which has been poured in. The measurement can be easily made by weighing the water, and the corresponding volume determined by calculation, making proper allowance for its temperature. The proportion required is the volume in cu. in. thus found, divided by the volume of the piston displacement, also in cu. in., and the result expressed as a decimal. In this test care should be taken that no air is retained in the clearance space when it is filled with water.

The only difficulty in measuring the clearance in this way is that occurring when the exhaust valves and piston are not tight, and the water poured in, flows away and is lost. If the leakage is serious, no satisfactory measurement can be made, and it is better to depend upon the volume calculated from the drawing. If not too serious, however, an allowance can be made by observing the length of time consumed in pouring in the water; then, after a portion of the water has leaked out, fill up the space again, taking the time and measuring the quantity thus added, determining in this way the rate of leakage. Data will thus be obtained for the desired correction.

APPENDIX NO. 2

LEAKAGE TESTS OF ENGINES, INCLUDING TIME METHOD

The method of testing the valves and pistons for leakage in a Corliss engine, or one in which the admission valves can be operated independently of the exhaust valves, is as follows:

First close the two steam valves, open the two indicator cocks, and admit a full pressure of steam into the chest by opening the throttle valve. The movement of the starting bar, first one way and then the other, so as to close one exhaust valve and then the other, causes the leakage through the steam valves to escape from the open indicator cock, where it becomes visible. The quantity of leakage is judged by the force of the current of steam blowing out.

To test the exhaust valves and piston, block the flywheel so that the piston will be at a short distance from the end of the stroke, and turn on the steam. The leakage escapes to the exhaust pipe, and can be observed at the open atmospheric outlet. If the outlet is not visible, and there is a valve in the exhaust pipe, this can be shut and the indicator cock opened. thereby deflecting the steam which leaks, and causing it to appear at the indicator cock. In a condensing engine where no atmospheric pipe is provided, and there is no opening that can be made in the exhaust pipe in front of the condenser, some idea can be obtained in regard to the amount of leakage by observing how rapidly the condenser is heated.

It is well to make these tests with the piston in different positions, so as to cover the whole range of the length of the stroke.

Another method of testing leakage is called the "time method." Instead of observing the steam that actually blows through the valves or piston to be tested, they are subjected to full steam pressure, and when the parts are thoroughly heated, the throttle valve is shut and the length of time observed which is required for the pressure to disappear. In testing the piston and exhaust valves, the flywheel is blocked as before, and, preferably, as indicator is attached, and a line drawn on a blank card at intervals of, say, one-quarter of a minute after the valve is shut, thereby making a record of the fall of the pressure. In a tight engine the fall of the pressure is slow, whereas in a leaky engine it is sometimes very rapid. The relative condition of the engine as compared with a tight engine must be judged by an observer, who must, of course, have had experience in tests of this kind on engines in various conditions.

The leakage of a piston can always be determined by removing the cylinder head and observing what blows through the open end with the pressure of steam behind it. The advantage of the time method is that it saves the labor and time required in removing the cylinder head and replacing it, which, in cases of large engines, is considerable.

Leakage tests of single-valve engines cannot be made as satisfactorily as those of the Corliss type and other four-valve engines. The best that can be done as regards the valve is to place it at or near the center of its travel, covering both ports, and then make the test under full pressure. The valve and piston can be tested as a whole by blocking the flywheel and opening the throttle valve in the same way as in other engines. In locomotives leakage will be revealed by the escape of steam at the top of the smoke stack.

In testing compound engines for leakage, the work is somewhat simplified in case of any one cylinder, as compared with a single engine. For example, leakage of the high-pressure cylinder can be revealed by opening the indicator cock on the proper end of the low-pressure cylinder, the steam valve of that cylinder being open. The test of leakage of the low-pressure exhaust valves and piston when the time method is used can be based on the indications of the receiver gage, instead of using an indicator. In that case the fall of the pressure due to leakage is read directly from the gage.

The tests thus far referred to are qualitative, and not quantitative. It is practical in some cases to determine the quantity of leakage under any set of conditions by collecting the steam which passes through, condensing it and weighing it. This can be readily done when there is a surface condenser, and it can be done in the absence of such a condenser by attaching a small pipe to the exhaust, and carrying the steam which escapes into a tank of water and condensing it. How much dependence can be placed upon the results of such a quantitative test as showing the actual quantity of leakage which occurs when the valves and pistons are in motion must be left to the judgment of the person who makes the test.

In Corliss engines the leakage of the piston with the engine in operation can be observed by removing the cylinder head, disconnecting the steam and exhaust valves at the head end, and setting the engine to work with steam admitted at the crank end.

APPENDIX NO. 3

GENERAL PRECAUTIONS REGARDING LEAKAGE AND METHODS OF MEASURING LEAKAGE

(a) *Leakage*

It is not always necessary to blank off a connecting pipe to make sure that there is no leakage through it. If satisfactory assurance can be had that there is no chance for leakage, this is sufficient. For example, where a straightway valve is used for cutting off a connecting pipe, and this valve has double seats with a hole in the bottom between them, this being provided with a plug or pet cock, assurance of the tightness of the valve when closed can be had by removing the plug or opening the cock. Likewise, if there is an open drip pipe attached to an unused or empty section of pipe beyond the valve, the fact that no water escapes here is sufficient evidence of the tightness of the valve. The main thing is to have positive evidence in regard to the tightness of the connections, such as may be obtained by the means suggested above; but where no positive evidence can be obtained, or where the leakage that occurs cannot be measured, it is of the utmost importance that the connections should be broken and blanked off.

Leakage of relief valves which are not tight, drips from traps, separators, etc., and leakage of tubes in the feedwater heater must all be guarded against or measured and allowed for.

It is well, as an additional precaution, to test the tightness of the feedwater pipes and apparatus concerned in the measurement of the water by running the pump at a slow speed for, say, fifteen minutes, having first shut the feed valves at the boilers and making sure they are tight. Leakage will be revealed by disappearance of water from the supply tank. In making this test, a gage should be placed on the pump discharge to guard against undue or dangerous pressure.

(b) *Water Glass Tests of Leakage*

To determine the leakage of steam and water from a boiler and steam pipes, etc., the water-glass method may be satisfactorily employed. This consists of shutting off all the feed valves (which must be known to be tight) and the main feed valve, thereby stopping absolutely the entrance or exit of water at the feed pipes to the boiler; then maintaining the steam pressure (by means of a very slow fire) at a fixed point, which is approximately that of the working pressure, and observing the rate at which the water falls in the gage glasses. It is well, in this test, as in other work of this character, to make observations every ten minutes, and to continue them for such length of time that the differences between successive readings attain a constant rate. In many cases the conditions will have become constant at the expiration of fifteen minutes from the time of shutting the valves, and thereafter the fall of water due to leakage of steam and water become approximately constant. It is usually sufficient, after this time, to continue the test for two hours, thereby obtaining a number of half-hourly periods. When this test is finished, the quantity of leakage is ascertained by calculating the volume of water which has disappeared, using the area of the water level and the depth shown on the glass, making due allowance for the weight of one cubic foot of water at the observed pressure. The water columns should not be blown down during the time a water-glass test is going on, nor for a period of at least one hour before it begins.

If there is opportunity for condensation to occur and collect in the steam pipe during the leakage test, the quantity should be determined as closely as desirable, and properly allowed for.

(*c*) *Surface Condenser Tests*

In making an engine test where the steam consumption is determined from the amount of water discharged from a surface condenser, leakage of the piston rods and valve rods should be guarded against; for if these are excessive, the test is of little use, as the leakage consists partly of steam that has already done work in the cylinder and of water condensed from the steam when in contact with the cylinder. If such leakage cannot be prevented, some allowance should be made for the quantity thus lost. The weight of water as shown at the condenser must be increased by the quantity allowed for this leakage.

APPENDIX NO. 4

CALIBRATING WATER METERS

Referring to Fig. 2, two tees *A* and *B* are placed in the feed pipe, and between them two valves *C* and *D*. The meter is connected between the outlets of the tees *A* and *B* and the valves *E* and *F* are placed one on each side of the meter. When the meter is running, the valves *E* and *F* are opened, and the valves *C* and *D* closed. A small bleeder *G* is kept open to make sure that there is no leakage. A gage is attached at *H*. When the meter is tested, the valves *C*, *D*, and *F* are closed, and the valves *E* and *I* are opened. The water flows from the valve *I* to a tank on platform scales. In testing the meter, the water is throttled at the valve *I* to obtain the desired rate of discharge, the gage meanwhile showing the working pressure. The piping leading from the valve *I* to the tank is arranged with a swinging joint, consisting merely of a loosely fitting elbow, so that it can be readily turned into the tank or away from it. When the desired speed has been secured, the end of the pipe is swung into the tank at the instant the pointer of the meter is opposite some graduation mark on the dial. When the required number of cubic feet are discharged, the pipe is swung away. The tests should start and stop at the same graduation mark on the first dial, and continued until at least 10 or 20 cu. ft. are discharged for one test. The tank is weighed before and after filling.

The water passing the meter should always be under pressure so that any air in the meter may be discharged through the vents provided for this purpose. Care should be taken that there is no unnecessary air drawn into the feed water. The meter should be tested before and after the trial, and repeated calibrations should be made to obtain confirmative results.

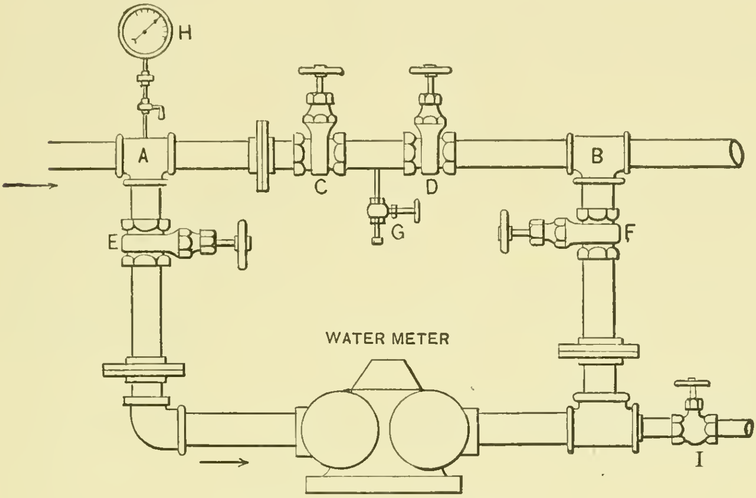

FIG. 2 METER CALIBRATION

Fig. 2* and the description apply to a piston meter, but any other type of meter carrying water under pressure may be calibrated in the same manner.

APPENDIX NO. 5

MEASUREMENTS OF WATER BY MEANS OF WEIRS, VENTURI METERS, ETC.

Weirs

The measurement of water by the use of weirs may be based on Francis' experiments, an account of which is given in Lowell's Hydraulic Experiments.[1] These resulted in the formula

$$Q = 3.33\ (L - 0.2H) \times H^{\frac{3}{2}}$$

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*Reproduced from Trans. Am. Soc. M. E., vol. 24, p. 724, fig. 118.

[1] D. Van Nostrand.

in which Q is the discharge in cu. ft. per second, L the length of the weir in ft., and H the depth of water on the weir in ft. The coefficient, 3.33 was obtained from the mean of 88 experiments, the greatest variation from the mean in any individual case being 1 per cent. The length of the weir. in all but six of the experiments, was approximately 10 ft. The depth of water on the weir varied from 7 to 19 in. The formula applies to that type of weir having perfect contraction at each end, which was the form used in 65 experiments.

The weir was of rectangular cross-section, with a horizontal crest and vertical ends. The upper edge was made of cast iron, and the corner presented to the current was square and sharp. The horizontal part of the crest was ¼ in. wide, and the remaining part was bevelled off at an angle of 45 deg. The ends were of similar cross-section to the crest. The depth on the weir was taken by means of hook gages, 6 ft. from the weir, these gages being placed in wooden boxes situated on the sides of the canal and communicating with the water through small openings. Vertical gratings. for overcoming eddies in the current, were provided above the gages. The distance from the side of the canal to the end of the weir was about 2 ft., and the depth of the canal below the crest was, in most of the experiments, about 5 ft.

The Francis formula is applicable only to cases similar to those described. According to Mr. Francis' statement, it cannot be applied where the depth on the weir exceeds one-third of the length, nor to very small depths. Furthermore, the distance from the side of the canal to the end of the weir should not be less than three times the depth on the weir.

In using the formula, the depth should be corrected for the head due to the velocity of approach, according to the formula

$$H' = \left[\left(H + h\right)^{\frac{3}{2}} - h^{\frac{3}{2}}\right]^{\frac{2}{3}}$$

in which H' is the corrected depth, H the observed depth, and h the head due to the velocity of approach. This last may be determined from the formula

$$h = \frac{V}{64.3}$$

in which V is the velocity of approach in ft. per second, which may be determined by dividing the uncorrected discharge of water, in cu. ft. per second, by the area of the cross-section of the steam flowing through the canal, in sq. ft.

Refer also to the experiments on weirs made by Hamilton Smith, Jr., described in Smith's Hydraulics, and to those made by Fteley and Stearns. described in the Transactions of the American Society of Civil Engineers, 1883.

Venturi Tubes

According to Herschel's experiments, described in the Transactions of the American Society of Civil Engineers, November 1887 and January 1888, a venturi tube inserted in a force main may be used for determining the quantity of water discharged by a pumping engine. Such a tube, applied. for example, to a 24-in. main, has a total length of about 20 ft. At a distance of 4 ft. from the end nearest the engine, the inside diameter of the tube is contracted to a throat having a diameter of about 8 in. A pressure gage is attached to each of two chambers, the one surrounding and communicating with the entrance, or main pipe, the other with the throat. Experiments made upon two tubes of this kind, one of which was 4 in. in diameter at the throat and 12 in. at its entrance, and the other about 36 in. in diameter at the throat and 9 ft. at its entrance, showed that the quantity of water which passes through the tube is very nearly the theoretical discharge through an opening having an area equal to that of the throat, and a velocity that due to the difference in head shown by the two gages. Mr. Herschel states that the coefficient for these two widely varying sizes of tubes and for a wide range of velocity through the pipe was found to be within 2 per cent, either way, of 98 per cent. In other words, the quantity of water flowing through the tube per second is expressed within 2 per cent by the formula

$$W = 0.98 \times A \times \sqrt{2gh}$$

in which A is the area of the throat of the tube, and h the head, in ft., corresponding to the difference in the pressure of the water entering the tube and that found at the throat.

The substantial reliability of venturi meters of small size (6 in. at entrance and 2 in. at throat), such as are readily calibrated, furnishes strong evidence of the practicability of this method of measurement for pumping engines.

Nozzles

The measurement of water by computation from its discharge through the nozzles of fire hose, furnishes a means of determining the quantity of water delivered by a pumping engine which can be applied without much difficulty. Freeman's investigations upon fire nozzles, described in the Transactions of the American Society Civil Engineers, November, 1889, which covered a wide range of pressures and sizes, showed that the coefficient of discharge for a smooth nozzle of ordinary good form was within ½ of 1 per cent. either way, of 0.977; the diameter of the nozzle being accurately calipered, and the pressures being determined by means of an accurate gage attached to a suitable piezometer at the base of the play pipe.

To use this method the water should be conducted to a pressure box and as many nozzles attached to the box as would be required to carry off the water. Four 1¼-in. nozzles, with a pressure of 80 lb. per sq. in., would discharge the full capacity of a 2,500,000 gal. engine.

Orifice Formula

The quantity of water flowing through an orifice into the open air, expressed in cu. ft. per second, may be calculated by the use of the formula

$$8.02\ AC\sqrt{H}$$

in which A is the area of the orifice in sq. ft., C, a coefficient depending on the form of the orifice and velocity of approach, and H, the head over the center of the orifice measured in ft. For an orifice located in the side of a tank, and consisting of a circular opening in a thin metal plate with a smooth sharp edge, the value of the coefficient is very close to 0.6. (See Hamilton Smith, Jr.'s, work on Hydraulics.)

Float Rods and Current Meters

The velocity of water flowing in a stream may be obtained indirectly by the use of float rods, and directly by means of current meters.

Float rods may be patterned after those used in Francis's experiments, which consisted of tin tubes, 2 in. in diameter, the lower ends being loaded so as to float upright, and the upper ends standing a few inches out of water. They were used at each foot of width of the canal, the length of run being 70 ft., and the time was recorded on a chronograph. The mean velocity ranged from 0.5 ft. to 5 ft. per second, the number of experiments being 115. It was found that if the lower ends of the rods did not extend to near the bottom of the canal, the observed velocity should be multiplied by a coefficient obtained from the formula

$$1 - 0.116\ (\sqrt{D} - 0.1)$$

in which D is the proportionate part of the total depth of the water not reached by the float. If for example the immersed part of the float is 0.9 of the total depth, D is 0.1 and the coefficient becomes $1 - 0.024 = 0.976$.

A current meter consists essentially of a small screw propeller which is operated by the current of flowing water in which it is immersed, the revolutions being shown on a dial placed at some convenient point to which the motion is transmitted electrically. The speed of revolution of the propeller, referred to a rating table, shows the velocity of the water. The Price electric meter is an example of a satisfactory instrument for this purpose. Whatever type of meter is used, its rating should be checked by calibration under conditions which are substantially the same as those occurring on the waterwheel test, especially those pertaining to depth of immersion and average velocity. The method of calibration considered most reliable is that which is carried on by moving the meter through still water at a known velocity.

For further information regarding these and other methods of measuring the velocity of water, reference may be made to Water Supply and Irrigation Paper No. 95, U. S. Geological Survey, published in 1904, on the subject, Accuracy of Stream Measurements.

APPENDIX NO. 6

PITOT TUBES FOR GAS AND AIR MEASUREMENT

A form of pitot tube suitable for measuring the velocity of gas or air moving in a pipe is illustrated in principle in Fig. 3. This consists of a small tube A, which is inserted in such a position that the open end points toward the current of gas or air, thereby receiving the full impact which the current produces. Surrounding this tube is an annular chamber or jacket B, which communicates with the outside space by means of four small holes C, D, E, and F, two on each side. By means of the small pipes G and H, the inner tube and the outer jacket are connected respectively to the two legs of a U-tube manometer, as shown in Fig. 4.

The pressure shown by the manometer, or what is called the velocity-pressure, is that due to the difference in pressure at the two points, the inner tube A being subjected to the sum of the static and velocity pressures, and the annular jacket B to the static pressure alone. The velocity corresponding to the difference of pressure shown is computed by using the formula

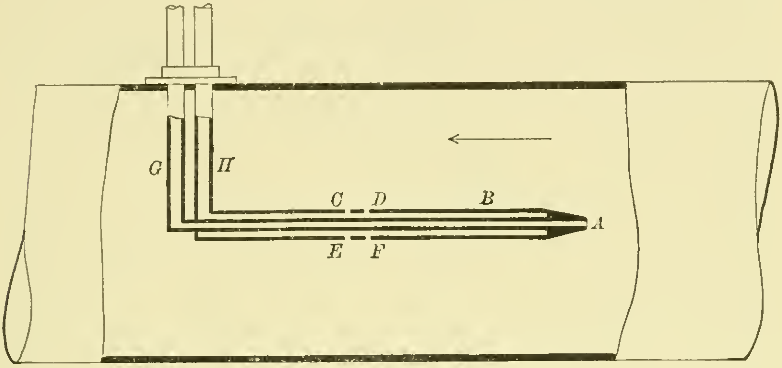

FIG. 3 PITOT TUBE

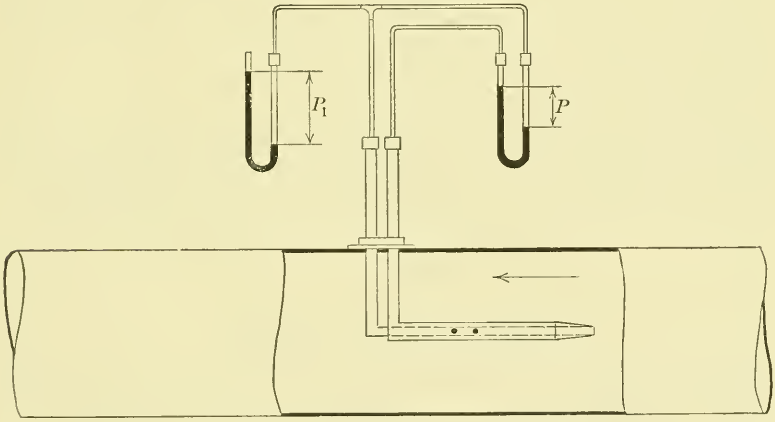

FIG. 4 PITOT TUBE AS INSTALLED

$$\text{Velocity per minute} = 60\sqrt{2\,gH}$$

in which $H = \dfrac{\text{Velocity-presssure in lb. per sq. ft.}}{\text{Weight of 1 cu. ft. of gas or air.}}$

The velocity-pressure in lb. per sq. ft. is found by multiplying the indication of the manometer P in in. of water by the constant 5.196. By substituting in the above formula, and reducing, we have

$$\text{Velocity per minute} = 1097\sqrt{\dfrac{\text{Reading of manometer in in. of water.}}{\text{Weight of 1 cu. ft. of gas or air.}}}$$

The weight of 1 cu. ft. of gas or air is obtained by multiplying the weight of 1 cu. ft. at a temperature of 32 deg. and pressure of 29.92 in. (0.0807 lb. for air) by the proportion obtained from the expression

$$\frac{492 \text{ deg.}}{460 + t} \times \frac{P + 29.92}{29.92}$$

in which t is the temperature of the gas or air as observed and P the observed pressure in in. of mercury.

For low velocities the U-tube shown in Fig. 4 may be replaced by one containing gasolene and having one leg inclined so as to multiply the ordinary indications to any extent desired. In this case the reading may be converted into inches of water by multiplying it by the proportion of the vertical height to the inclined height and by the specific gravity of gasolene.

For high pressures, the manometer attached to the impact tube can be replaced by a mercury column or steam gage attached to a receiver in the main where the velocity is largely reduced.

To determine the average velocity of the gas or air throughout the whole cross-section of the pipe, readings should be obtained at a number of points and the results averaged. Two sets of readings should be taken, one set along the horizontal diameter and the other set on the vertical diameter. The various points should be selected so that the different velocities will conform to equal areas. If there are ten points in each diameter this equality will be obtained if they are located on the four radii at distances of 32, 55, 71, 84, and 95 per cent, respectively, of their length, measured from the center. In pipes carrying high pressures two pitot tubes may be used, one for each diameter, and they should be mounted so as to be readily moved to the desired locations, the required points being indicated by suitable marks on the outside supports.

The point selected for the attachment of a pitot tube should be as far removed as practicable from abrupt bends.

Another form of tube may be used, when properly calibrated, especially for high pressures, which is inserted diametrically across the main in the manner of a sampling pipe. In this case the impact is obtained by a number of short tubes attached to the side and pointing toward the current.

APPENDIX NO. 7

BITUMINOUS COAL SIZES

Bituminous coals in the Eastern States may be graded and sized as follows:

(*A*) Run of mine coal; the unscreened coal taken from the mine.

(*B*) Lump coal; that which passes over a bar-screen with openings 1¼-in. wide.

(*C*) Nut coal; that which passes through a bar-screen with 1¼-in. openings and over one with ¾-in. openings.

(*D*) Slack coal; that which passes through a bar-screen with ¾-in. openings.

Bituminous coals in the Western States may be graded and sized as follows:

(*E*) Run of mine coal; the unscreened coal taken from the mine.

(*F*) Lump coal; divided into 6-in., 3-in. and 1¼ in. lump, according to the diameter of the circular openings over which the respective grades pass; also 6 by 3 lump and 3 by 1¼ lump, according as the coal passes through a circular opening having the diameter of the larger figure and over that of the smaller diameter.

(*G*) Nut coal; divided into 3-in. steam nut, which passes through an opening 3-in. diameter and over 1¼-in.; 1¼-in. nut, which passes through a 1¼-in. diameter opening and over a ¾-in. diameter opening; ¾-in. nut, which passes through a ¾-in. diameter opening and over a ⅝-in. diameter opening.

(*H*) Screenings; that which passes through a 1¼-in. diameter opening.

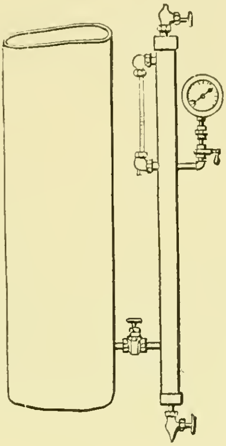

Fig. 5 Air Chamber for Attachment of Gage to Force Main

(*I*) Washed sizes; those passing through or over the circular openings of the following diameters, in inches:

Number	Through	Over
1	3	1¾
2	1¾	1⅛
3	1⅛	¾
4	¾	¼
5	¼	...

APPENDIX NO. 8

APPARATUS FOR PUMPING ENGINES

To determine the length of stroke in the case of direct-acting engines, a scale should be securely fastened to the frame which connects the steam and water cylinders, in a position parallel to the piston rod, and a pointer attached to the rod so as to move back and forth over the graduations on the scale. The marks on the scale, which the pointer reaches at the two ends of the stroke, are thus readily observed, and the distance moved over computed. If the length of the stroke can be determined by the use of some form of registering apparatus, such a method of measurement is preferred. The personal errors in observing the exact scale marks, which are liable to creep in, may thereby be avoided.

The apparatus referred to under the heading (*g*) *Pressure Gages*, page 1694, consisting of a small air chamber for the attachment of a gage to the force main is shown in Fig. 5*.

APPENDIX NO. 9

HUMIDITY TABLE

Table 6 is condensed from the hygrometric tables of U. S. Government referred to on Page 1696.

APPENDIX NO. 10

OPTICAL, PNEUMATIC, AND RADIATION PYROMETERS

Wanner's optical pyrometer, Uehling's pneumatic pyrometer, and Ferry's radiation pyrometer, are among those referred to on page 1696 as being useful for measuring high temperatures. Full descriptions of these instruments may be found in the catalogues of various dealers in chemical and physical apparatus.

APPENDIX NO. 11

DESCRIPTION OF STEAM CALORIMETERS

The instrument referred to on page 1697 is shown in Figs. 6*a* and 6*b*. It consists of brass pipe and fittings of the ½-in. size in which the leading parts are the two tees, a pair of flanges, an orifice plate between the flanges, and two thermometer cups containing thermometers which are screwed into the tees. The whole is surrounded by suitable non-conducting covering and encased in a box. The percentage of moisture is determined by observing the number of degrees of cooling that the thermometer in the low-pressure steam shows the "normal" reading for dry steam, and dividing that number by the "constant" number of degrees representing 1 per cent of moisture.

To determine the "normal" reading of the low-pressure thermometer corresponding to dry steam, the instrument should be attached to a horizontal steam pipe in such a way that the sampling nozzle projects upwards to near the top of the pipe, there being no perforations and the steam entering through the open top of the nozzle. The test should be made when the steam in the pipe is in a quiescent state, and when the steam pressure is maintained constantly at the point observed on the main trial. If the steam pressure falls during the time when the observations are being made, the test should be continued long enough to obtain the effect of an equivalent rise of pressure.

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* Reproduced from Trans. Am. Soc. M. E., vol. 12, p. 518, fig. 109.

To find the "constant" for 1 per cent of moisture divide the latent heat of the steam supplied to the calorimeter at the observed pressure or temperature by the specific heat of superheated steam at atmospheric pressure (0.46) and divide the quotient by 100.

TABLE 6. RELATIVE HUMIDITY, PER CENT.

Dry Thermometer, Deg. Fahr.	Difference between the Dry and Wet Thermometers, Deg. Fahr.													
	1	2	3	4	5	6	7	8	9	10	11	12	13	14
	Relative Humidity, Saturation being 100 (Barometer = 30 in.)													
32	89	79	69	59	49	39	30	20	11	2	..	..	..	..
40	92	83	75	68	60	52	45	37	29	23	15	7	0	..
50	93	87	80	74	67	61	55	49	43	38	32	27	21	16
60	94	89	83	78	73	68	63	58	53	48	43	39	34	30
70	95	90	86	81	77	72	68	64	59	55	51	48	44	40
80	96	91	87	83	79	75	72	68	64	61	57	54	50	47
90	96	92	89	85	81	78	74	71	68	65	61	58	55	52
100	96	93	89	86	83	80	77	73	70	68	65	62	59	56
110	97	93	90	87	84	81	78	75	73	70	67	65	62	60
120	97	94	91	88	85	82	80	77	74	72	69	67	65	62
140	97	95	92	89	87	84	82	79	77	75	73	70	68	66

	15	16	17	18	19	20	21	22	23	24	26	28	30
32	..	..	..	..	..	..	..	..	..	..	..	..	..
40	..	..	..	..	..	..	..	..	..	..	..	..	..
50	11	5	0	..	..	..	..	..	..	..	..	..	..
60	26	21	17	13	9	5	1	..	..	..	..	..	..
70	36	33	29	25	22	19	15	12	9	6	..	..	..
80	44	41	38	35	29	26	23	20	18	12	7	..	..
90	49	47	44	41	39	36	34	31	29	26	22	17	13
100	54	51	49	46	44	41	39	37	35	33	28	24	21
110	57	55	52	50	48	46	44	42	40	38	34	30	26
120	60	58	55	53	51	49	47	45	43	41	38	34	31
140	64	62	60	58	56	54	53	51	49	47	44	41	38

Finally ascertain the percentage of moisture by dividing the number of degrees of cooling by the constant, as above noted.

To determine the quantity of steam used by the calorimeter it is usually sufficient to calculate the quantity from the area of the orifice and the absolute pressure, using Napier's formula for the number of lb. which passes through per second; that is, absolute pressure in lb. per sq. in. divided by 70 and multiplied by the area of orifice in sq. in. To determine the quantity by actual test, a steam hose may be attached to the outlet of the calorimeter, and carried to a barrel of water on platform scales. The

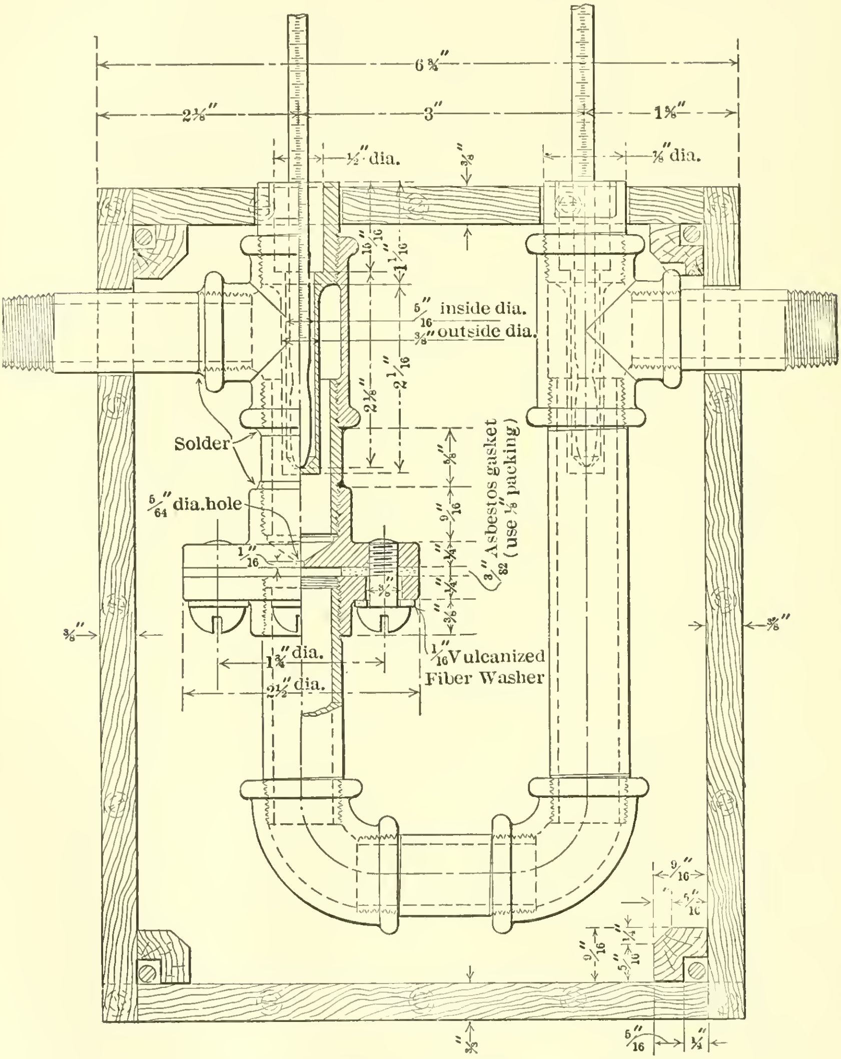

FIG. 6*a* THROTTLING CALORIMETER

amount of steam condensed in a certain time is determined, and thereby the quantity discharged per hour.

A combined throttling and separating calorimeter, in which the throttling portion operates on the same principle as the one just described, is shown in Volume 17 of the Transactions.[1]

APPENDIX NO. 12

MAHLER COAL CALORIMETER

The Mahler coal calorimeter consists essentially of a strong cylindrical vessel having a capacity of about 800 cu. c., which is closed at the top and filled with oxygen gas, under a pressure of 375 lb. per sq. in. A sample of finely powdered coal, which will pass through a sieve having 10,000 meshes to the sq. in., weighing about 1 gram, is placed in a pan suspended within the interior vessel, and provided with two electrodes, through which an electric current from a battery can be passed. The whole is immersed in an outer vessel containing about 2500 grams of water, thoroughly stirred, the temperature of the water observed, the coal set on fire by completing the electric circuit, the water again stirred, and the temperature observed at intervals of half a minute until the thermometer ceases to rise. The difference between the initial and final temperatures thus determined is corrercted for radiation, the latter being found by observing the rate at which the temperature changes before and after the coal is fired. The weight of water contained in the outer vessel, the water equivalent of the whole apparatus, and the weight of coal being known, the total heat of combustion is found by multiplying the sum of the two water weights in grams by the corrected rise of temperature in deg. fahr., and the product is divided by the weight of the coal in grams.

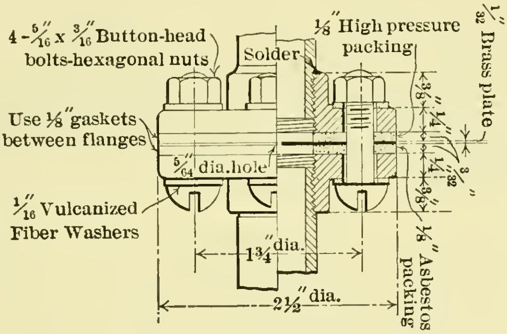
4-5/16" x 3/16" Button-head bolts-hexagonal nuts
Solder
1/8" High pressure packing
1/32" Brass plate
Use 1/8" gaskets between flanges
5/64" dia. hole
1/16" Vulcanized Fiber Washers
3/8"
1/4"
1/4"
3/32"
3/8"
1¾" dia.
2½" dia.
1/8" Asbestos packing

Fig. 6b Orifice Plate for Throttling Calorimeter

The sample used for the calorimeter test should be air dried. A similar air-dried sample should be tested for moisture in the manner described at the end of page 1712, the final result being based on dry coal.

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[1] R. C. Carpenter. New Form of Steam Calorimeter. Trans. Am. Soc. M. E., vol. 17, p. 618.

APPENDIX NO. 13

JUNKER GAS CALORIMETER

This instrument consists of a vertical cylindrical water chamber containing vertical tubes, which is heated by the gas burned in a Bunsen lamp beneath. The products of combustion pass upward through the tubes, and the water flows downward through the chamber in a continuous current. The quantity of gas is measured by a gas meter, and the quantity of water by collecting the overflow discharged from the apparatus. Thermometers are inserted at the points of entrance and exit. The heat of combustion of a cu. ft. of gas is determined by multiplying the rise of temperature in deg. fahr. by the weight of water in lb., and dividing the product by the volume of gas in cu. ft. The result thus found is what is termed the "higher value," and this is the value, unless otherwise stated, which is employed throughout the codes.

The "low value" is obtained by multiplying the weight of the condensed vapor resulting from the combustion, expressed in lb., by the total heat of atmospheric steam above the temperature of the condensed vapor, dividing the product by the volume of the gas in cu. ft., and subtracting the quotient from the higher value.

APPENDIX NO. 14

FUEL ANALYSES

Proximate Analysis of Coal

The apparatus required for proximate analysis consists of a mill for grinding coal, chemical scales sensitive to 1/1000 of the amount weighed, drying apparatus, a platinum crucible, a Bunsen burner and blast lamp, a supply of oxygen gas, and such chemicals and chemical apparatus as may be required. The elements to be determined are moisture, volatile matter, fixed carbon, ash, and sulphur.

Determine the moisture as explained at the end of Part III *A*.

To determine volatile matter, place one gram of air-dried powdered coal in the crucible and cover it with a loose platinum plate. Heat 3½ minutes in the flame of the Bunsen burner, and continue the heating for 3½ minutes longer in the flame of the blast lamp. Cool down, remove the cover, and weigh the residue. The loss in weight represents the combined volatile matter and moisture. Subtracting the moisture, the weight of volatile matter alone is determined.

To ascertain the ash, expose the residue in the crucible to the blast lamp until it is completely burned, using a stream of oxygen if desired to hasten the process. The residue left is the ash.

The difference between the residue left after the expulsion of the volatile matter and the ash is the fixed carbon.

To determine sulphur by Eschka's method, which is the one commonly used, heat 1 gram of coal mixed with 1 gram of magnesium oxide and ½ gram of sodium carbonate for 1 hour, using an alcohol lamp. After cooling, mix with 1 gram of ammonium nitrate and heat the mixture ten minutes; then dissolve in 200 cc. of water, heat and reduce by evaporation to 150 cc., acidify with hydrochloric acid, and filter. Add barium chloride to the filtrate and determine the sulphur by calculation from the quantity and composition of the barium sulphate thereby precipitated.

Ultimate Analysis of Coal

The apparatus required for ultimate analysis consists of a mill and other apparatus for grinding and pulverizing the coal; chemical scales sensitive to 1/1000 of the amount weighed; drying apparatus; combustion apparatus, embracing a combustion furnace, a glass combustion tube, one end of which is filled with copper oxide and chromate of lead and the other end with a roll of oxidized copper gauze, a porcelain boat, a set of bulbs containing hydrate of potassium, a U-tube filled with chloride of calcium, and a supply of pure oxygen and pure air; together with suitable chemicals and chemical apparatus required for the various processes. The elements to be determined are moisture, carbon, hydrogen, oxygen, sulphur, nitrogen, and ash.

The moisture is determined in the manner pointed out under Proximate Analysis.

The carbon and hydrogen are obtained by the use of the combustion apparatus. One-half gram of the pulverized air-dried coal is placed in the porcelain boat, which is introduced between the copper roll and the copper oxide within the combustion tube. After the coal and the entire contents within have been thoroughly dried out by a sufficient preliminary heating, aided by a current of dry air, the furnace is set to work and the coal burned by first passing air through the tube and finally oxygen, conducting the products of combustion through the potash bulbs and the chloride of calcium tube. The carbon dioxide produced by the combustion of the carbon is absorbed by the potash, and the water formed by the combustion of hydrogen, together with that due to the moisture in the air-dried coal, is taken up by the chloride of calcium. The quantity of carbon is determined by weighing the bulbs before and after, thereby obtaining the weight of the carbon dioxide produced, and then calculating the weight of carbon from the known composition of the dioxide. Likewise, the quantity of hydrogen is determined by weighing the calcium tube before and after, which, after deducting the moisture in the air-dried coal, gives the amount of water produced, and, dividing by 9, the amount of hydrogen.

Sulphur is found by the method described above under the heading Proximate Analysis.

To determine nitrogen, a certain weight of coal is mixed with strong sulphuric acid and permanganate of potash and heated until nearly colorless. This process converts the nitrogen into ammonia and then into sulphate of ammonia, and the amount of sulphate is determined by making the solution alkaline and then distilling it. The nitrogen is found by calculation from the known composition of ammonia.

The ash is found by weighing the refuse left in the combustion boat after the coal is completely burned.

The oxygen is the difference between the sum of the elements previously determined and the original weight of coal.

The ultimate analysis of coal, as will be seen from the above description, requires the use of so much chemical apparatus, and at best it is so complicated, that it is not likely to be done except in a fully equipped chemical laboratory. It should not be undertaken by one who is not entirely familiar with all the details of the work.

Analysis of Liquid Fuels

The determination of carbon and hydrogen in liquid fuels is made in the same manner as that concerning the solid fuels above described, using special means for preventing loss in the various processes on account of the volatile characteristics of the fuel.

To determine the sulphur, the oil or other liquid is heated with nitric acid and barium chloride. The quantity of sulphate of barium thus produced is ascertained by filtering and weighing, and the sulphur calculated from the known composition of the compound.

The ultimate analysis of liquid fuel, like that of coal, should be undertaken only by a person familiar with all the necessary details.

See report of Committee of American Chemical Society on Coal Sampling and Analysis.

APPENDIX NO. 15

GAS ANALYSIS

Orsat Apparatus

The Orsat apparatus is a portable instrument contained in a wooden case with removable sliding doors front and back, as shown in its simplest form in Fig. 7*. It consists essentially of a measuring tube or burette, three absorbing bottles or pipettes, and a leveling bottle, together with the connecting tubes and apparatus. The bottle and measuring tube contain pure water; the first pipette, sodium hydrate dissolved in three times its weight of water; the second, pyrogallic acid dissolved in sodium hydrate in the proportion of 5 grams of the acid to 100 cc. of the hydrate; and the third, cuprous chloride. These chemicals are sold by most of the large dealers.

The manipulation of the instrument, which can be carried on after suitable practice by any person familiar with testing work, is as follows:

After completely drawing out the air contained in the supply pipe, a sample of the gas is drawn into the measuring tube by opening the necessary connections and allowing the water to empty itself from the tube and flow into the bottle. The quantity of gas drawn in is adjusted to 100 cc. By opening one by one the connections to the pipettes, and raising and lowering the water bottle, the sample is alternately admitted to and withdrawn from the pipettes, and the ingredients one by one absorbed.

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* Reproduced from Trans. Am. Soc. M. E., vol. 18, p. 903, figs. 290, 291.

The first pipette absorbs carbon dioxide (CO_2) ; the second. oxygen (O) ; and the third. carbonic oxide (CO). The quantity absorbed in each case is determined by finally returning the sample to the measuring burette and reading the volume. The percentage of CO_2 is read directly, being the first absorption. That of the other two ingredients are the respective differences between the readings taken after successive absorptions.

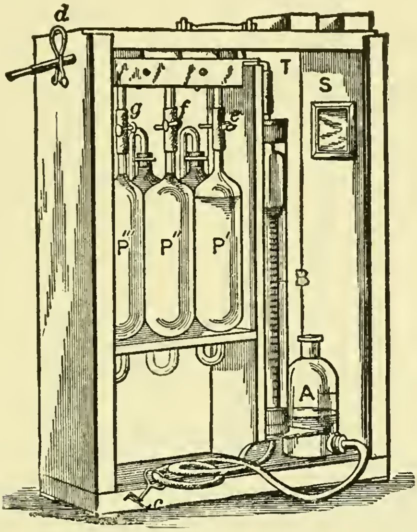

FIG. 7 ORSAT APPARATUS

Various modifications of this apparatus have been developed which enable analyses to be made with greater rapidity than with the form illustrated in Fig. 7.

Hempel Apparatus

The Hempel apparatus works on the same principle as the simple form of Orsat apparatus described, so far as the latter is applicable, excepting that the absorption may be hastened by shaking the pipettes bodily, bringing the chemical into most intimate contact with the gas. It is less portable and in some particulars it requires more careful manipulation than the Orsat, while for general analysis it is not adapted unless used in a well equipped chemical laboratory. The absorption pipettes are made in sets which are shaped in the form of globes, and a number of independent sets are required for the treatment of the different constituent gases. A simple pipette of the Hempel type is shown in Fig. 8.

The method of carrying on an analysis with the Hempel apparatus is as follows:

> A sample of gas measuring 100 cc. is drawn into the burette, and then transferred to the first pipette, which contains potassium hydrate dissolved in two times its weight of water. This pipette absorbs carbon dioxide (CO_2). The gas is then passed into the second pipette, containing saturated bromine water, which absorbs the heavy hydrocarbons (C_2H_4); then into the third pipette, containing a solution of pyrogallic acid and potassium hydrate in the proportion of 5 grams of acid to 100 cc. of hydrate, which absorbs oxygen (O); then into the fourth pipette, containing ammoniacal cuprous chloride, which absorbs carbonic oxide (CO), and finally into the fifth pipette, which is of large size and provided with exploding wires and galvanic battery, for the determination of marsh gas (CH_4) and hydrogen (H). A measured quantity of oxygen gas is added to this pipette and the contents exploded by an electric spark from the battery, resulting in a mixture of carbon dioxide, nitrogen, and free oxygen. The quantity of carbon dioxide is determined by passing the gas into the pipette containing potassium hydrate, and the quantity of oxygen by subsequently passing it into the pipette containing potassium pyrogallate, finally determining the quantity of marsh gas and hydrogen from the known reactions which occur during this process, and the composition of the resulting gases.

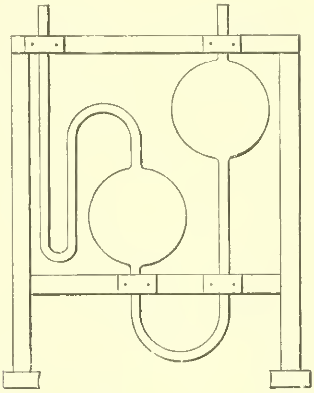

FIG. 8 HEMPEL PIPETTE

For each of these processes the pipettes are shaken to hasten the absorption, and the quantity absorbed is determined by returning the gas into the measuring burette and observing the successive differences.

Tar and Soot.

The quantity of tar and soot contained in producer gas may be found by drawing a measured volume of gas through filter paper, weighing the paper before and after.

APPENDIX NO. 16

RINGELMANN CHART

A Ringelmann smoke chart is shown with full-size spacing in Fig. 9*. To use this chart, four cards are ruled like those shown, though covering a much larger area, and placed in a horizontal row about 50 ft. from the observer, and in line between him and the chimney, together with two other cards, one of which is white and the other solid black. The observer glances rapidly from the chimney to the cards and judges which one corresponds with the color and density of the smoke. He makes these observations every minute, or oftener if desired, recording the number of the card representing the character of the smoke at the instant of observation. The results are then plotted on a chart, and the variations shown graphically.

The lines in cards 1 to 4 are respectively 1, 2.3, 3.7, and 5.5 mm. thick, and the spaces 9, 7.7, 6.3, and 4.5 mm.

APPENDIX NO. 17

SMOKE METER

As an illustration of the soot-collecting method referred to on page 1699, which was carried out in a certain trial, a plate ⅞ in. wide and 24 in. long, representing a surface of 21 sq. in., was inserted through a hole in the top of the flue and suspended by a wire, the hole being covered after the plate was inserted. The plate was temporarily withdrawn every two hours during the progress of the test, and the collection of soot removed by a brush. This was tried on a boiler fitted with an automatic stoker and on another fitted with a smoke-burning furnace, and it was found that under various conditions as to character of coal and other changes, the weight of soot collected per hour varied from 9 to 184 mg.

APPENDIX NO. 18

USE AND CALIBRATION OF INDICATORS

The indicated horsepower should be determined from the average mean effective pressure of diagrams taken at intervals of 20 minutes, and at more frequent intervals if the nature of the test makes this necessary, this being done for each end of each cylinder. With variable loads, such as

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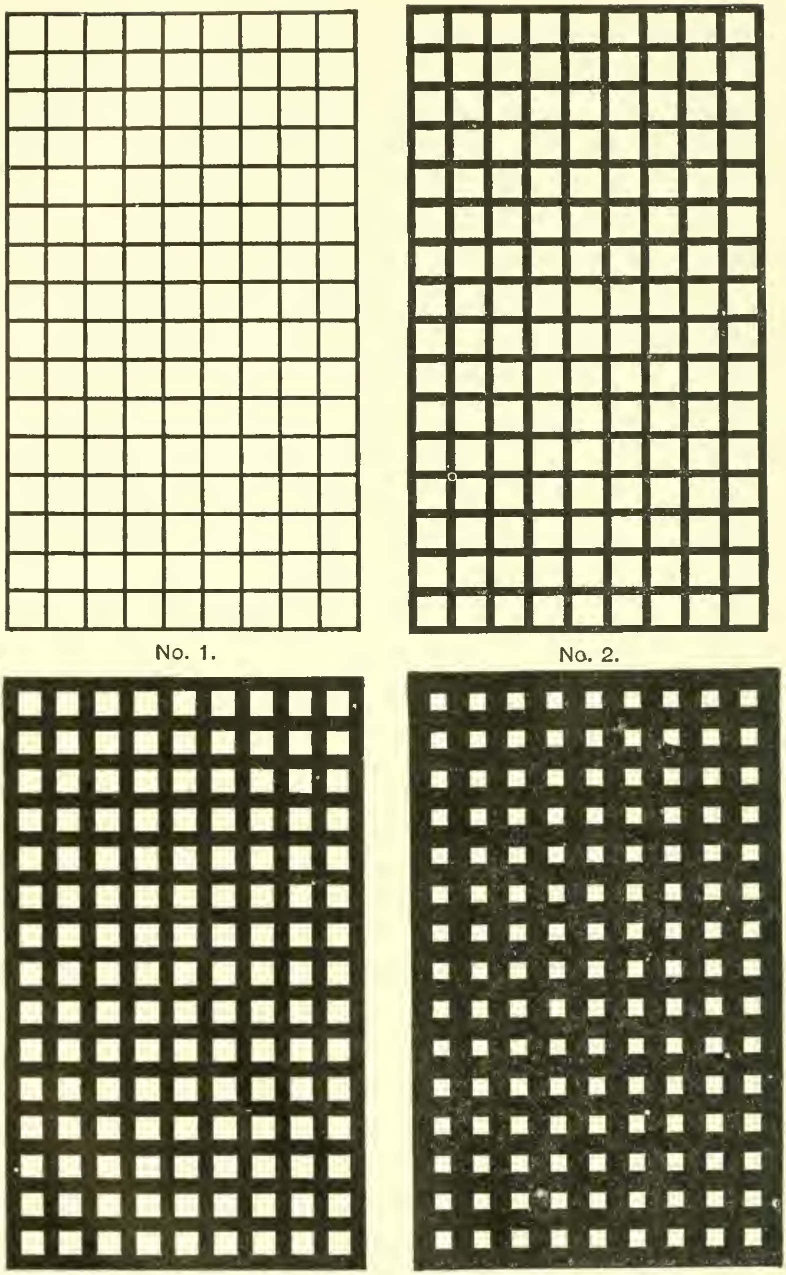
* Reproduced from Trans. Am. Soc. M. E., vol. 21, p. 98, fig. 6.



No. 1. No. 2.

No. 3. No. 4.

Fig. 9 Ringelmann Smoke Chart

those of engines driving generators for electric railroad work, and of rubber-grinding and rolling-mill engines, the diagrams cannot be taken too often. In cases like the latter, one method of obtaining suitable averages is to take a series of diagrams on the same blank card without unhooking the driving cord, and apply the pencil at successive intervals of 10 seconds until 2 minutes' time or more has elapsed, thereby obtaining a dozen or more indications in the time covered. This tends to insure the determination of a fair average for that period. In taking diagrams for variable loads, as indeed for any load, the pencil should be applied long enough to cover several successive revolutions, so that the variations produced by the action of the governor may be properly recorded. To determine whether the governor is subject to what is called "racing" or "hunting," a variation diagram should be obtained; that is, one in which the pencil is applied a sufficient time to cover a complete cycle of variations. When the governor is found to be working in this manner, the defect should be remedied before proceeding with the test.

It is seldom necessary, as far as average power measurements are concerned, to obtain diagrams at precisely the same instant at the two ends of the cylinder, or at the same instant on all the cylinders, when there are more than one. All that is required is to take the diagram at regular intervals. Should the diagrams vary so much among themselves that the average may not be a fair one, it signifies that they should be taken more frequently, and not that special care should be employed to obtain the diagrams of each set at precisely the same time. When diagrams are taken during the time when the engine is working up to speed at the start, or when a study of valve setting and steam distribution is being made, they should be taken at as nearly the same instant as practicable. In cases where the diagrams are to be taken simultaneously, the best plan is to have an operator stationed at each indicator. This is desirable, even where an electric or other device is employed to operate all the instruments at once; for unless there are enough operators, it is necessary to open the indicator cocks some time before taking the diagrams and run the risk of clogging the pistons and heating the high-pressure springs above the ordinary working temperature.

To determine the power developed while an engine is starting from rest and attaining working speed and working load, a number of diagrams should be taken in rapid succession, the speed counted for each one, and the results averaged. If the period of time thus covered on a commercial test is unduly long so as to be an appreciable percentage of the total running time, this power should be included in the average for the whole run, due regard being had for the time it is in operation. In that event the total duration of the run should be the whole time the throttle valve is open.

The most satisfactory driving rig for indicating is some form of well-made pantagraph, with driving cord of fine annealed wire leading to the indicator. The reducing motion, whatever it may be, and the connections to the indicator, should be so perfect as to produce diagrams of equal length when the same indicator is attached to either end of the cylinder, and produce a proportionate reduction of the motion of the piston at every point of the stroke, as proved by test.

With a perfect working pantagraph, or similar apparatus, the equality in the length of diagrams taken with the same indicator at the two ends is sufficient indication of the substantial reliability of the reduction when the point of cut-off on the diagram is not unusually short, say, not shorter than one-eighth. When the cut-off is unusually short, the error produced by imperfect reducing motion, stretch of cord, or otherwise, becomes a comparatively large item, and one which for accurate work should be allowed for. To test the accuracy of the reducing motion without making special preparations for a thorough examination, it is sufficient to make a comparison between the actual proportion of the stroke covered and the apparent proportion measured on the indicator, and see how they agree. This may be done on a large engine by making the comparison wherever it happens to stop, and repeating the comparison when it has stopped with the piston at some other point of the stroke. With an engine which can be turned over by hand, or where auxiliary power is provided for moving it, the comparison may be made at a number of equi-distant points in the stroke. To make the test properly, a diagram should be taken just before stopping, and this will serve as a reference for the measurements taken after stopping. The actual proportion of stroke covered is determined by measuring the distance which the crosshead has moved and comparing it with the whole length of the stroke, making sure that the slack has all been taken up by turning sufficient steam into the cylinder to bring a pressure to bear on the piston, but not sufficient to start the flywheel in motion. To obtain the apparent indication from the diagram, the indicator pencil is moved up and down with the finger, so as to make a vertical mark on the diagram, and the distance of this mark from the beginning of the diagram compared to the whole length is the proportion desired.

It is necessary, of course, to go through these operations without changing in any way the adjustment of the driving cord of the indicator, or any part of the mechanism that would alter the movements of the indicator.

In all cases the pipes leading to indicators should be as short and direct as possible. The use of a three-way cock and a single indicator connected to the two ends of the cylinder is not advised, except in cases where it is impracticable to use an indicator close to each end. If a three-way cock is used, the error produced should be determined and allowed for.

The effect of the error produced by a three-way cock is usually to increase the area of the diagram. This is due to the tardiness of the indicator in responding to the changes of pressure. In an investigation which was carried out both on short-stroke engines running at high speed and long-stroke engines running at comparatively slow speed, it was found that the increased area of the diagram, due to the sluggish action referred to, ranged from 3 to 7 per cent as compared with an indicator having a short and direct pipe.

The point selected for attaching the indicator pipe to the cylinder should never be the drip pipe or any point where the water of condensation will run into the instrument, if this can possibly be avoided. The admission of water with the steam may greatly distort the diagram. If it becomes necessary to place the indicator in such a position, as may happen when it is attached to the lower end of a vertical cylinder, the connection to the indicator must be short and direct, and in some cases it should be provided with a drip chamber arranged so as to collect the water or deflect it from entering the instrument.

The pipe connections for indicating gas and oil engines should be removed as far as possible from the ports and ignition devices, and made preferably in the cylinder head. The pipes should be as short and direct as possible. Avoid the use of long pipes, otherwise explosions of the gas in these connections may occur.

Ordinary indicators suitable for indicating steam engines are much too lightly constructed for gas and oil engines. The pencil mechanism, especially the pencil arm, needs to be very strong to prevent injury by the sudden impact at the instant of explosion. A special gas-engine indicator is required for satisfactory work, with a small piston and a strong spring.

In the manipulation of the indicator it is important to keep the instrument in clean condition and preserve it in mechanically good order. Ordinary cylinder oil is the best material to use for lubricating the indicator piston for pressures above the atmosphere. It is better to have the piston fit the cylinder rather loosely, so as to get absolute freedom of motion.

than to have a mathematically accurate fit. In the latter case, extreme care and frequent cleanings are required to obtain good diagrams.

> No diagrams should be accepted in which there is any appearance of want of freedom in the movement of the mechanism. A ragged or serrated line in the region of the expansion or compression lines is a sure indication that the piston or some part of the mechanism sticks: and when this state of things is revealed the indicator should not be trusted, but the cause should be ascertained and a suitable remedy applied. Entire absence of wire drawing of the steam line, and especially a sharp, square corner at the beginning of the steam line, should be looked upon with suspicion, however desirable and satisfactory these features might otherwise be. These are frequently produced by an indicator which is defective owing to want of freedom in the mechanism. An indicator which is free when subjected to a steady steam pressure, as it is under a test of the springs for calibration, should be able to produce the same horizontal line, or substantially the same, after pushing the pencil down with the finger, as that traced after pushing the pencil up and subsequently tapping it lightly. When the pencil is moved by the finger, first up and then down, the piston being subjected to pressure, the movement should appear smooth to the sense of feeling.

To make a comparison of indicator springs with standards, the calibration should be made, if this were practical, under the same conditions as those pertaining to their ordinary use. Owing to the fact that for steam work the pressure of the steam in the indicator cylinder and the corresponding temperature are undergoing continual changes, it becomes almost impossible to compare the springs with any standard under such conditions. There must be a constant pressure during the time that the comparison is being made. To bring the conditions for steam work as nearly as possible to those of the working indicator, the steam should be admitted to the indicator as short a time as practicable for each of the pressures tried, and then the indicator cock should be closed and the steam exhausted therefrom before another pressure is tried. By this means the parts are heated and cooled somewhat the same as under the working conditions. For each required pressure the first step is to open and close the indicator cock a number of times in quick succession, then quickly draw the line for the desired record, observing the gage or other standard at the same instant. A corresponding atmospheric line is taken immediately afterwards.

The calibration should be made for at least five equidistant points. For ordinary work the arithmetical mean of the various results should be taken for the average scale.

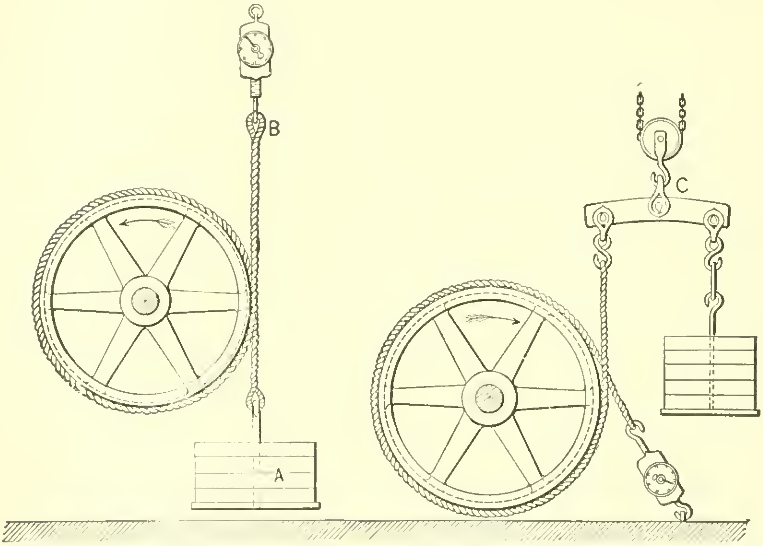
The indicator springs used for gas and oil engines should also be calibrated with the indicator in as nearly as possible the same condition as to temperature as exists during the trial. A simple way of heating the indicator is to subject it to a steam pressure just before calibration. Compressed air is suitable for the actual work of calibration, being used in preference to steam so as to bring the conditions as near as possible to those which obtain when the indicators are in actual use in such engines.

The standard of comparison in all cases is a steam gage which has been proved correct by reference to either a mercury column or a dead weight testing apparatus.

APPENDIX NO. 19

FRICTION BRAKES

A self-adjusting rope brake is illustrated in Fig. 10* in which it will be seen that if the friction at the rim of the wheel increases, it will lift the weight *A*, which action will diminish the tension in the end *B* of the rope, and thus prevent a further increase in the friction. The same device can be used for a band brake of the ordinary construction. Where space below the wheel is limited, a cross bar, *C*, supported by a chain tackle exactly at its central point, may be used as shown in Fig. 11*, thereby causing the action of the weight on the brake to be upward. A safety stop should be used with either form, to prevent the weights being accidentally raised more than a certain amount.



Figs. 10 and 11 Rope Brakes

A water-friction brake is shown in Fig. 12*. It consists of two circular disks, *A* and *B*, attached to the shaft *C*, and revolving in a case *E*, between fixed planes. The space between the disks and planes is supplied with running water, which enters at *D* and escapes at the cocks *F*, *G*, and *H*. The friction of the water against the surfaces constitutes a resistance which absorbs the desired power, and the heat generated within is carried away by the water itself. The water is thrown outward by centrifugal action and fills the outer portion of the case. The greater the depth of the ring of water, the greater the amount of power absorbed. By adjusting suitably the amount of water entering and leaving any desired power can be obtained.

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* Reproduced from Trans. Am. Soc. M. E., vol. 24, pp. 739 and 740, figs. 119, 120, 121.

Another form of water brake is that shown in Fig. 13, which is used by the Westinghouse Machine Company for large turbines. In this brake the shaft is carried in two bearings and is coupled to the turbine shaft. The revolving paddle wheel or "runner" is designed to run in very close clearance with the serrated-rim piece. The outer casing of the brake maintains its accurate position surrounding the runner by means of carefully constructed bearings. To this casing is rigidly attached the lever arm, made of such length as to facilitate ready calculation. The little roller wheel on the end of the arm bears upon platform scales which weigh the load.

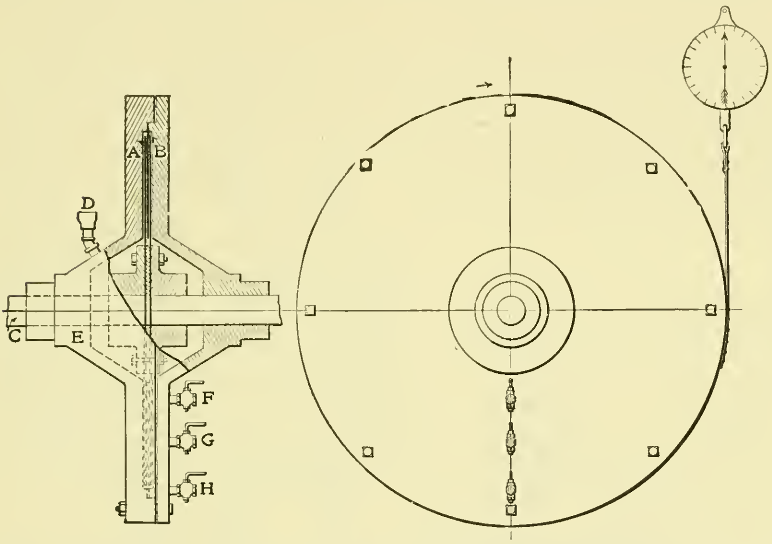

Fig. 12 Water Brake

Water is introduced through a hose connection at the opening marked *B*. The water enters the interior of the runner, is thrown out by centrifugal force through small holes drilled in the rim into the outer teeth spaces where a resistance is created, and it escapes with difficulty through the close clearance around the outside of the runner, finally discharging at the bottom through openings *C* (Fig. 12) and *C* (Fig. 13), where it runs to waste.

For a description of the Alden water brake, see the paper on An Automatic Absorption Dynamometer.[1]

The application of water brakes for making field tests of waterwheels is shown in Professor Allen's paper on The Testing of Water Wheels After Installation.[2]

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[1] Geo. I. Alden. Trans. Am. Soc. M. E., vol. 11, p. 958.

[2] C. M. Allen. Trans. Am. Soc. M. E., vol. 32, p. 275.

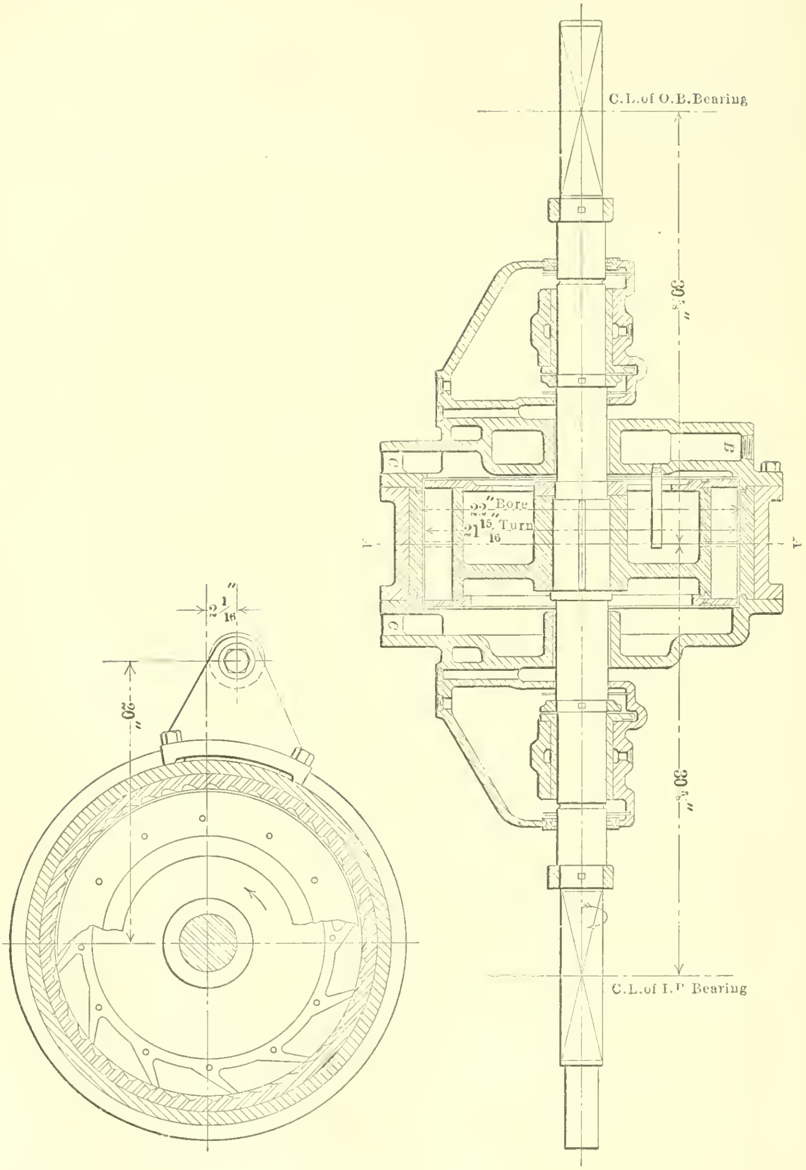

FIG. 13 WATER BRAKE

APPENDIX NO. 20

DYNAMOMETERS

A form of dynamometer for measuring the power transmitted from an engine to a mill shaft is well illustrated in principle by the Webber balance dynamometer, shown in Vol. 4 of the Transactions.[1] In this apparatus, which consists essentially of a compound gear, the intermediate gear is carried by a weighted arm, which is free to rise and fall, according to the amount of force transmitted. The power is determined from the net weight thus lifted.

Another form which is useful for mill purposes is the belt dynamometer, the principle of which is shown in Fig. 14. *A* is the driving pulley and *B* the driven pulley. The connecting belt passes over the dynamometer pulleys *C* and *D*. These two pulleys are mounted on a frame carrying a scale beam *F*, all of which turns on the center *E*. The difference in the total strain on the two sides of the belt is computed by multiplying the net weight on the beam by the leverage, the latter being found by dividing the length of the beam by the distance from the center *E* to the centers of pulleys *C* and *D*.

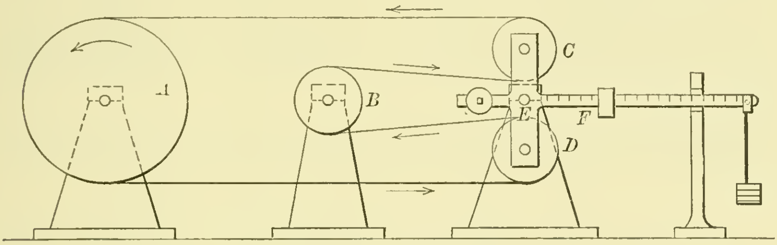

FIG. 14 BELT DYNAMOMETER

The *shaft dynamometer* consists of the following essential parts: a long tube encircling the shaft and made fast thereto at one end, being free at the other end and maintained in alignment by adjustable rollers; two radial arms, one attached to the shaft and the other to the free end of the tube, which rotate a slight amount with reference to each other, according to the twist of the enclosed length of shaft; and a set of levers which multiply this rotative movement and at the same time convert it into linear motion, which is transmitted to a sleeve and collar mounted upon the shaft and sliding thereon. These parts all revolve with the shaft. An independent indicating apparatus is provided, which is mounted on a stationary frame, and the sliding movement of the rotating collar is transmitted through it to an index hand. The torsional strain is determined from the reading of the accompanying scale, which is graduated to millimeters.

The zero reading is found by disconnecting the propeller and turning the shaft at a slow speed, first in one direction and then in the other, observing the indication in both cases, and fixing the point of zero strain at the mean of the two. When it is impracticable to disconnect the propeller the readings may be taken when the vessel is drifting under her own headway after shutting off steam. The calibration of the instrument, which can best be done when the shaft is in the shop before installation, is carried on by securing the shaft in a fixed position, and applying a torsional strain by means of weights at the end of a lever attached beyond the dynamometer, taking readings with a number of different weights.

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[1] S. S. and W. O. Webber. Trans. Am. Soc. M. E., vol. 4, p. 227.

The horsepower shown by the dynamometer is determined by multiplying the reading of the instrument expressed in millimeters by the number of revolutions of the shaft per minute, and by a constant determined from the calibration noted, the constant being an expression for the horsepower corresponding to 1 r.p.m. and a reading of 1 mm.

A shaft dynamometer requires delicate manipulation, a load of 500 h.p. in some instruments causing a movement at the end of the two arms of only 1/100 of an inch.

The *locomotive dynamometer*, as formerly designed, consists essentially of one or more strong helical springs placed between the drawbar of the locomotive and the train, and the distance which the spring is extended furnishes a measurement on an appropriate scale of the amount of force transmitted, and thereby the amount of power. The strain on the spring is transmitted through mechanism to a registering apparatus placed in the dynamometer car, and a continuous record is made upon a strip of paper traveling at a definite rate of speed. The apparatus used by the Pennsylvania R. R., the Chicago, Burlington and Quincy R. R., and the Chicago, Milwaukee and St. Paul R. R., are shown in the cuts given on pages 1329 to 1334 of Transactions, Vol. 14.

The latest form of dynamometer car makes use of oil plungers in place of springs, and employs oil transmission for the recording devices.

APPENDIX NO. 21

USE AND CALIBRATION OF ELECTRICAL INSTRUMENTS

To determine whether the readings of any instrument are disturbed by stray fields, the instrument is turned bodily through an angle of 180 deg. If there are stray fields, the change of position will change the reading; in which case the instrument should be moved to some other point where the reading is found to be unaffected.

The proper methods of connecting wattmeters in polyphase alternating-current systems are shown in Figs. 15 and 16. The former refers to two-phase and the latter to three-phase systems. In Fig. 15 there are four cables, designated *A*, *B*, *C*, and *D*, arranged in two pairs, or one pair for each phase. Each wattmeter is connected to its respective pair of cables, the current coil being in series with one of the two cables, and the potential coil in parallel with the load or bridging the two cables. In Fig. 16 there are three cables, which for convenience are designated by the letters *X*, *Y*, and *Z*. Meter W_1 is connected to the phase represented by the cables *X* and *Y*, and meter W_2 to that represented by the cables *Y* and *Z*. The current coil of the former is connected in series with cable *X*, the potential coil bridging *X* and *Y*. The current coil of the latter is connected in series with cable *Z*, the potential coil bridging *Y* and *Z*. The connections shown are elementary; transformers and other equipment usually employed in actual installations being omitted. The total output is the arithmetical sum of the quantities indicated by the two meters for a two-phase system, and the algebraic sum for a three-phase system.

The power factor is the proportion borne by the true power or kilowatt output as shown by wattmeters, to the apparent power or kilovolt-ampere (kv. a.) output determined from the readings of the voltmeters and ammeters. In a single-phase system the kv.a. output is found by multiplying the volts by the amperes and dividing by 1000. In a two-phase system lt is found by multiplying the volts by the amperes in each phase, adding the two products together, and dividing by 1000. In a three-phase system it is ascertained by multiplying the average volts of the three circuits by the average amperes of the three circuits, multiplying the product by the constant 1.73, and dividing by 1000.

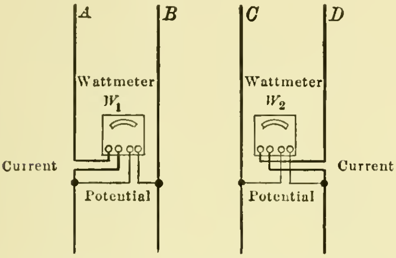

Fig. 15 Two-Phase System

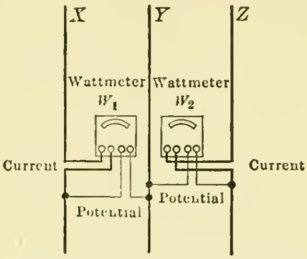

Fig. 16 Three-Phase System

In a three-phase system which is balanced (that is, when there is an equality of amperes and volts in all three phases), the power factor may also be determined by referring to the curve shown in Fig. 17, in which abscissae represent the proportion borne by the readings of the two wattmeters, and ordinates the corresponding power factor.

When the power factors in a three-phase system is 100 per cent, both wattmeters indicate the same quantity. When it is less than 100 per cent, one of the meters indicates a larger quantity than the other, the one showing the larger quantity giving positive readings, and the other either positive or negative readings. The latter become negative when the power factor falls below 50 per cent. To determine whether the smaller reading is to be considered negative or not, disconnect the potential circuit of the meter showing the smaller reading from the cable *Y* (referring to Fig. 16), and connect it to the opposite outside cable so that this circuit will bridge the two outside cables. If the meter reverses, i.e., if the index hand falls back to the zero point and rests against the stop, the readings taken with the meter in its working position must be considered negative. If it does not reverse, the readings are both positive.

A watt-hour meter can best be calibrated by reference to an indicating wattmeter, in accordance with the following directions:

Ascertain first the constants and formula pertaining to the watthour meter by referring to the dial of the instrument, and, if need be, to the manufacturer's instruction book; and by this means determine the number of watts corresponding to one revolution of the meter disk per second. Then find the actual speed of the disk by observing the time required for a certain number of complete revolutions, using a stop watch, and compute the actual number of revolutions per second; finally, multiplying the revolutions per second by the watts corresponding to one revolution per second, and the result is the output in watts. Read the indicating wattmeter every five seconds during the period that the speed is observed, and average the readings. A comparison of the watts determined by the watt-hour meter observations with this average shows the error of the watt-hour meter.

Calibrations should be made while the main test is going on. If the load is not substantially constant they should be repeated at regular intervals during its progress so as to obtain the comparison under average conditions.

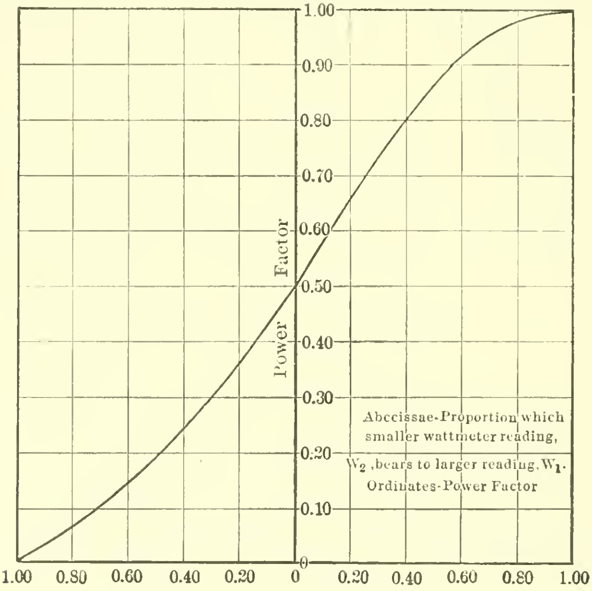

Fig. 17 Power Factor Curve

Reference may be made to the Standardization Rules of the American Institute of Electrical Engineers for methods of determining generator temperatures and for general information bearing on electrical standards.

APPENDIX NO. 22

WATER RHEOSTATS

A simple form of water-rheostat, suitable for absorbing the output of a 500-kw. generator of the three-phase alternating-current type, consists of three wine casks, one for each phase, placed side by side, through

APPENDIX NO. 23

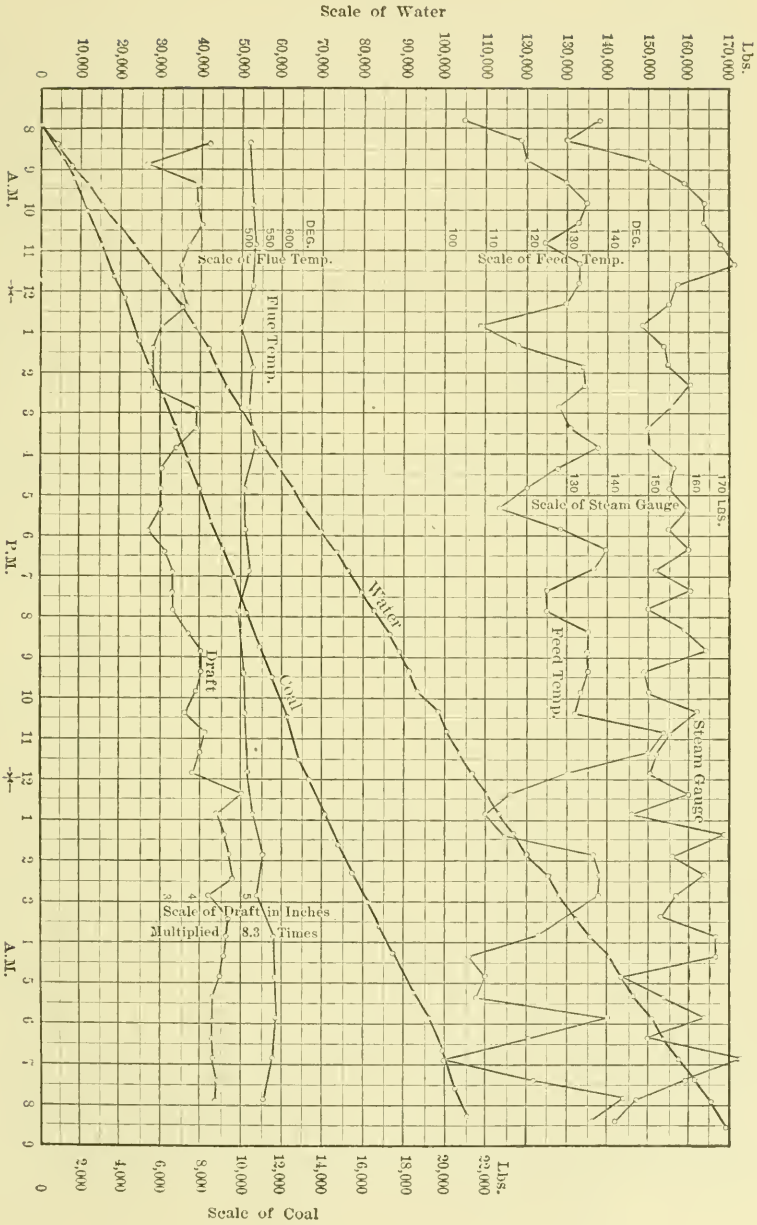

FIG. 18. CHART SHOWING LOG OF BOILER TEST

Reproduced from Trans. Am. Soc. M. E. vol. 21, p. 104, fig 7.

each of which a sufficient amount of cooling water is allowed to flow to prevent boiling; salt being added to the water as may be needed to lower its resistance. A fixed terminal is provided consisting of three connected iron plates, one of which is laid on the bottom of each cask. A movable terminal is provided for each phase, consisting of a piece of 2-in. standard iron pipe, 4 to 6 ft. in length, suspended vertically in the water at the center of each cask. Each pipe is arranged to be raised or lowered at will, so as to regulate the depth to which it is immersed, and thereby the output for that phase. By varying the relative immersion of the pipes, an equal distribution of the current may be obtained between the three phases.

Another arrangement consists of a tank, the size of which equals that of the three casks combined, the pipes being located a suitable distance apart and a single plate being used for the fixed terminal.

When a larger amount of current than 500 kw. is generated, the size of the tank and terminals may be proportionately increased, and at the same time provision made for introducing the cooling water so as to flow along the surface of each pipe and overcome or reduce the tendency to local ebullition.

The changes of resistance due to variations in the strength of the salt solution, depth of immersion of the terminals, and temperature of the water, are such that constant attention is required to maintain uniformity in the output with this kind of rheostat. When uniformity is desired without such attention a rheostat may be constructed which embodies the principles referred to in the description of the apparatus used for a direct current rheostat, which is given on Page 1701.

APPENDIX NO. 24

LOCATION OF INSTRUMENTS FOR BOILER TESTS

(*a*) The feedwater thermometer should be placed in a thermometer well and inserted in the feed pipe. Except in cases where an injector is used, the point selected should be as near as practicable to the boiler. Where an injector is employed, and the water is weighed or measured before it is supplied thereto, the well should be placed on the suction side of the injector, and the injector should receive steam through a short covered pipe connected directly to the boiler under test. If the steam is taken from some other source and it is of different pressure and different quality from that of the boiler under test, correction should be made for such difference, and especially for any excessive moisture thus introduced into the feedwater. When the temperature of the water changes between the injector and boiler, as by the use of a heater or by excessive radiation, the temperature at which the water not only enters and leaves the injector, but that also at which it enters the boiler, should all be taken. In that case, the weight to be used is that of the water leaving the injector, computed from the heat units if not directly measured and the temperature, that of the water entering the boiler. The weight of condensed steam to be added to the weight of water entering the injector, to obtain that leaving the injector, may be computed by multiplying the weight entering by the proportion

$$\frac{h_3 - h_1}{h_2 - h_3}$$

in which

h_1 = heat units per lb. of water entering injector
h_2 = heat units per lb. of steam entering injector
h_3 = heat units per lb. of water leaving injector

(*b*) The location of the steam calorimeter and steam thermometer should be as close to the boiler as possible, keeping in mind the directions given on page 1713.

(*c*) Draft gages should be attached to each boiler between the hand damper and the boiler, and as near the damper as practicable. In the case of a plant containing a number of boilers, a gage should also be attached to the main flue between the regulating damper and the boiler plant. It is desirable also to have gages connected to the furnace or furnaces of the boilers, and in cases of forced blast, to the ashpits and blower ducts. If there is an economizer in the flue a gage should be connected to the flue at each end of this apparatus. The same draft gage may be used for all the points noted, provided suitable pipes are run from the gage to each, arranged so as to be readily connected to either point at will.

(*d*) The flue thermometer should be located where it will show the average temperature of the whole body of gas. For an extremely large flue the thermometer may be placed in an oil pot of small diameter, which is suspended in the flue, and the thermometer lifted partially out of the oil when the temperature is read.

(*e*) The sample for flue gas analysis should be drawn from the region near the center of the main body of escaping gases, and the point selected should be one where there is no chance for air leakage into the flue, which could affect the average quality. In a round or square flue having an area of not more than one-eighth of the grate surface, the sampling pipe may be introduced horizontally at a central point, or preferably a little higher than the central point, and the pipe should contain perforations extending the whole length of the part immersed and pointing toward the current of gas, the collective area of the perforations being less than the area of the pipe.

APPENDIX NO. 25

DETERMINATION OF HEAT CONSUMPTION OF ENGINES AND TURBINES

The measurement of the heat consumption requires the measurement of each supply of feedwater to the boiler—that is, the water supplied by the main feed pump, that supplied by auxiliary pumps, such as jacket water, water from separators, drips, etc., and water supplied by gravity or other means; also the determination of the temperature of the water supplied from each source, together with the pressure and quality of the steam.

The temperatures at the various points should be those applying to the working conditions. It frequently happens that the measurement of the water requires a change in the usual temperature of supply. For example, where the main supply is ordinarily drawn from a hot-well in which the temperature is, say 100 deg. fahr., it may be necessary, owing to the low level of the well, to take the supply from some source under a pressure or head sufficient to fill the weighing tanks used, and this supply may have a temperature much below that of the hot-well; possibly as low as 40 deg. fahr. The temperature to be used is not the temperature of the water as weighed in this case, but that of the working temperature of the hot-well. The working temperature in cases like this must be determined by a special test, and included in the log sheets.

In determining the working temperatures, the preliminary or subsequent test should be continued a sufficient time to obtain uniform indications, and such as may be judged to be an average for the working conditions. In this test it is necessary to have some guide as to the quantity of work being done, and for this reason the power developed by the engine should be determined by obtaining a full set of diagrams at suitable intervals during the progress of the trial, and in a turbo-generator, the output of the generator should be observed. Observations should also be made of the gages connected with the plant and of the water levels in the boilers, the latter being maintained at a uniform point so as to be sure that the rate of feeding during the test is not sensibly different from that of the main test.

When the feedwater is all supplied by one feeding instrument, the temperature to be found is that of the water in the feed pipe near the point where it enters the boiler. If the water is fed by an injector this temperature is to be corrected for the heat added to the water by the injector, and for this purpose the temperatures of the water entering and of that leaving the injector are both observed. If the water does not pass through a heater on its way to the boiler (that is, that form of heater which depends upon the rejecting heat of the engine or turbine, such as that contained in the exhaust steam either of the main cylinders or turbine or of the auxiliary pumps) it is sufficient, for practical purposes, to take the temperature of the water at the source of supply, whether the feeding instrument is a pump or an injector.

When there are two independent sources of feedwater supply, one the main supply from the hot-well, or from some other source, and the other an auxiliary supply derived say from the water condensed in the jackets of the main engine and in the live-steam reheater, if one be used, they are to be treated independently. The remarks already made apply to the first, or main, supply. The temperature of the auxiliary supply, if carried by an independent pipe either direct to the boiler or to the main feed pipe near the boiler, is to be taken at a convenient point in the independent pipe.

When a separator is used in the main steam pipe, arranged so as to discharge the entrained water back into the boiler by gravity, no account need be made of the temperature of the water thus returned. Should it discharge either into the atmosphere to waste, to the hot-well, or to the jacket tank, its temperature is to be determined at the point where the water leaves the separator before its pressure is reduced.

When a separator is used, and it drains by gravity into the jacket tank, this tank being subjected to boiler pressure, the temperatures of the separator water and jacket water are each to be taken before their entrance to the tank.

The heat to be determined is that used by the entire engine or turbine equipment, embracing the main cylinders or turbine and all auxiliary cylinders and mechanism concerned in the operation of the engine, including the air pump, circulating pump, and feed pumps, also the jacket and reheater when these are used. No deduction is to be made for steam used by auxiliaries unless these are shown by test to be unduly wasteful. In this matter an exception should be made in cases of guarantee tests where the engine contractor furnishes all the auxiliaries referred to. He should, in that case, be responsible for the whole, and no allowance should be made for inferior economy, if such exists. Should a deduction be made on account of the auxiliaries being unduly wasteful, the method of waste and its extent, as compared with a standard of known value, should be definitely noted.

The steam pressure and the quality of the steam are to be taken at some point conveniently near the throttle valve. The quantity of steam used by the calorimeter must be determined and properly allowed for.

APPENDIX NO. 26

BRITISH STANDARD OF ENGINE ECONOMY

The British standard, which gives the heat consumption per minute per unit of power, relates to the heat calculated from an assumed temperature of the feedwater, which is taken as that corresponding to the back pressure steam, allowance being made for heat derived from the drips of jackets, reheaters, etc.

As to the adaptability of the British standard, as compared with the standard given in this code, it should be borne in mind that the practical utility of an engine depends almost wholly upon the fact that it is used for some form of industrial work, and, in combination with a boiler and various appurtenances, it is a part of a complete power plant. Were it not for the unreliable character of coal and other fuels, the proper standard of economy for such an engine, with a boiler of given efficiency, would be the fuel consumed, because fuel is, in reality, the source of the power, and this is the most important thing to be supplied in operating the engine. The use of a standard of heat units in place of fuel, meets the objectionable characteristics of coal due to its heat variability, and in doing this it makes no change in the conditions under which the standard should be employed. These are the actual conditions of use, and not the ideal conditions. The British method furnishes a means of determining and stating the ideal performance of an engine working under assumed conditions. It does not give the performance of the entire engine plant under the actual conditions as affected by the efficiency of the heaters and auxiliaries, although the performance under such conditions is usually of the greatest importance.

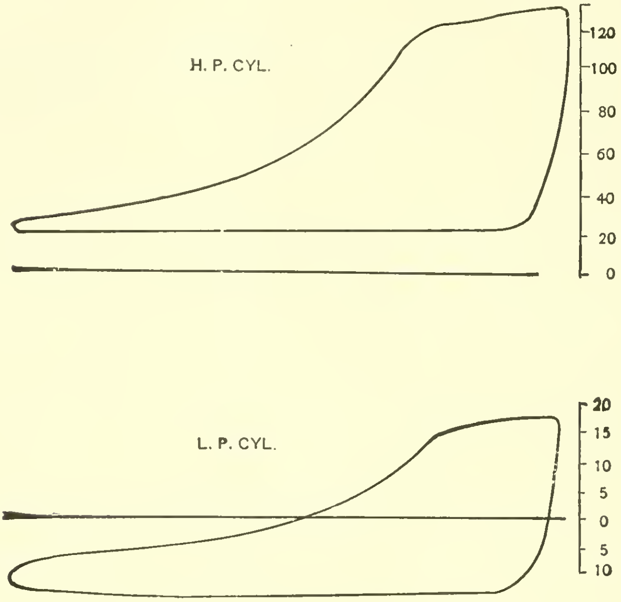

Fig. 19 Actual Diagram Receiver Engine

The British standard is useful in the following particulars:

(1) It expresses the economy of the engine as an independent machine, unaffected by the failings of the feedwater heater or condenser.

(2) In this form, it is useful for expressing the economy of an engine in the testing shop of an engine builder, or in the mechanical laboratory of a college, where it forms no part of a power plant.

(3) A comparison of the British standard with the economy of an ideal engine working between the same limits of temperature reveals those losses going on which are due to the engine alone, and this furnishes information not otherwise obtained.

Note: A method of studying cylinder performance by means of logarithmic diagrams has been proposed by J. Paul Clayton. See The Journal, Am. Soc. M. E., April 1912.

APPENDIX NO. 27

THE COMBINED DIAGRAM

The Combined Diagram is a hypothetical figure, which in its essential features represents an indicator diagram which would be obtained if the whole process of admission, expansion and exhaust occurred in one cylinder, viz., the low-pressure cylinder. It is a diagram from which the pressure of the steam at any point in the stroke of either cylinder, and the volume of that steam can be measured from one diagram, in the same manner that it can be measured in the case of a single cylinder engine from the actual indicator diagram.

The general method of laying out a combined diagram is shown clearly in Figs. 19 to 22*, the first of which refers to a Corliss compound engine (receiver type) in which the cylinder ratio is 3.72, and the clearance 4 per cent and 8 per cent, respectively, and the second to a Westinghouse compound engine (Woolf type), in which the cylinder ratio is 2.72, and the clearances. 33 per cent and 9 per cent, respectively.

APPENDIX NO. 28

DIAGRAM FACTOR

The diagram factor is the proportion borne by the mean effective pressure measured from the actual diagram to that of an ideal diagram which represents the maximum power obtainable from the steam accounted for by the actual diagram at the point of cut-off; assuming first that the engine has no clearance; second, that there are no losses through wire-drawing the steam either during admission or release; third, that the expansion line is a hyperbolic curve; and, fourth, that the initial pressure is that of the boiler; and the back pressure that of the atmosphere for a non-condensing engine, and of the condenser for a condensing engine.

The method of determining the diagram factor is illustrated in Figs. 23 to 26*, which apply to a simple non-condensing engine, a simple condensing engine, and a compound condensing engine. In Fig. 23, *RS* represents the volume of steam at boiler pressure admitted to the cylinder, *PR* and *OS* being hyperbolic curves drawn through the compression and cut-off points respectively. In Fig. 24 the factor is the proportion borne by the area of the actual diagram to that of the diagram *CNHSK*. In Fig. 25 the factor is the proportion borne to the arc of the diagram *CNHSK*. In Fig. 26 the factor is the proportion borne by the area of the two combined diagrams to the area *CNHSK*. In Fig. 24 where the diagram is the same as in Fig. 23, the distance *CN* is laid off equal to *RS* shown in Fig. 23, and the curve *NH* is a hyperbola referred to the zero lines *CM* and *MJ*. In Fig. 25 the distance *CN* is found in a similar way. In Fig. 26 the distance *CN* for the high-pressure cylinder is found in the same manner as in the case of a simple engine. The mean effective pressure of the ideal diagram can readily be obtained from the formula

$$\frac{P}{R}\ (1 + \text{hyperbolic logarithm } R) - p$$

where

P = absolute pressure of steam in boiler

$R = \dfrac{MJ}{CN}$

p = pressure of atmosphere or that in condenser

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* Reproduced from Trans. Am. Soc. M. E., vol. 24, pp. 746, 748, 753, 754, figs. 123-126, 130-133.

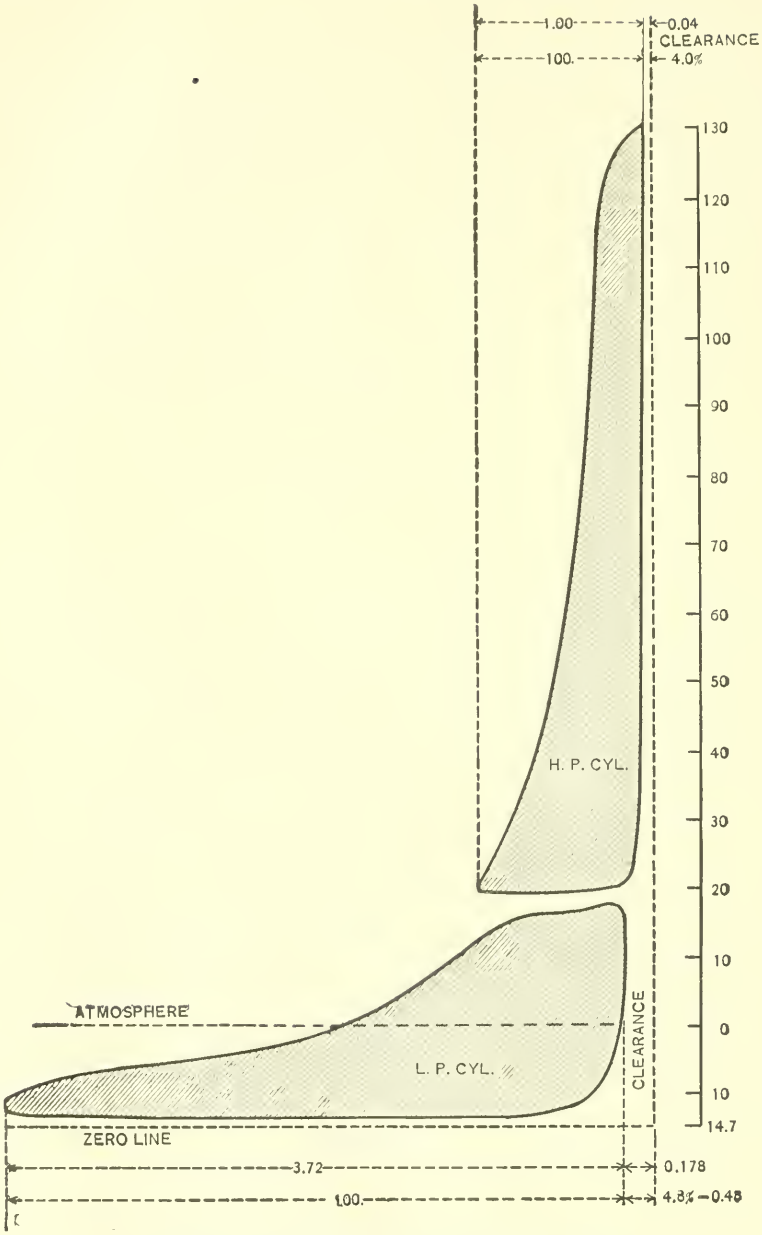

Fig. 20 Combined Diagram Receiver Engine

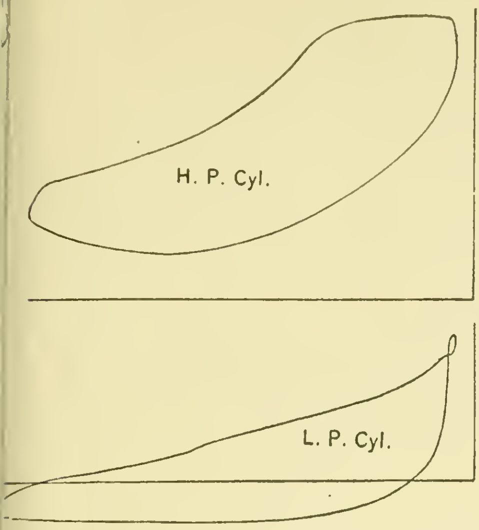

FIG. 21 ACTUAL DIAGRAM WOOLF ENGINE

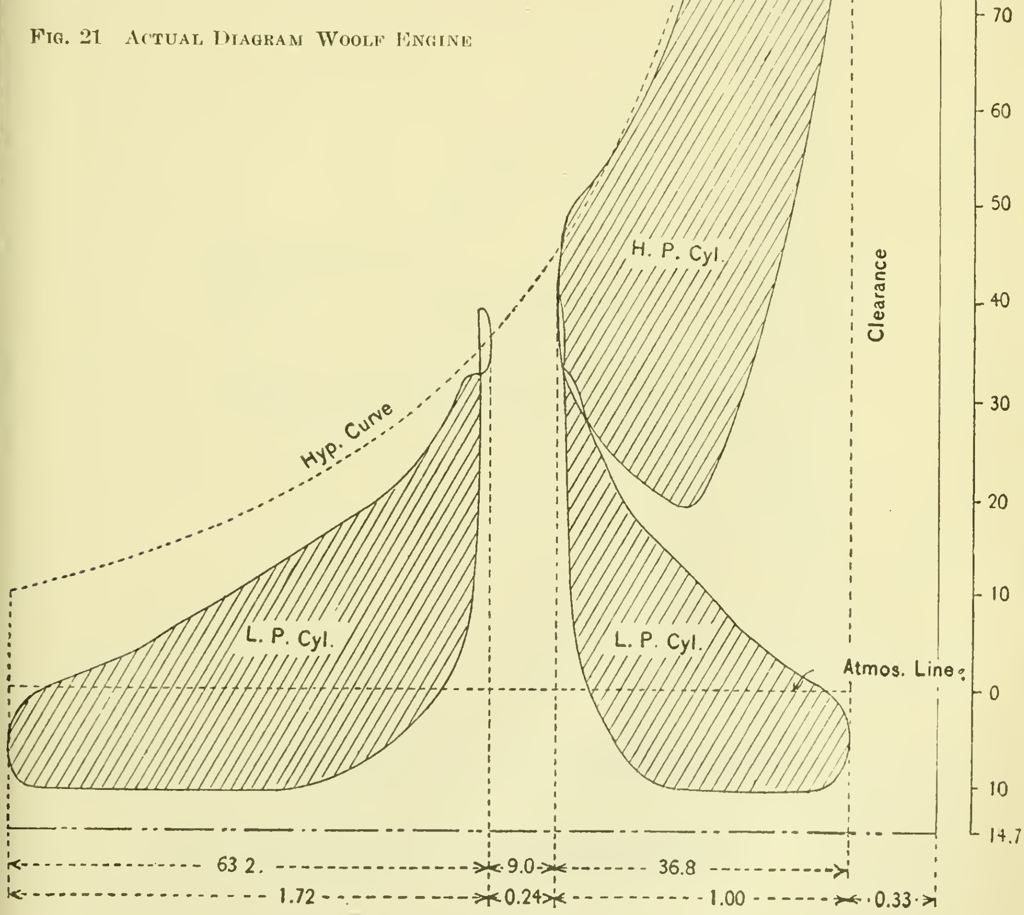

FIG. 22 COMBINED DIAGRAM WOOLF ENGINE

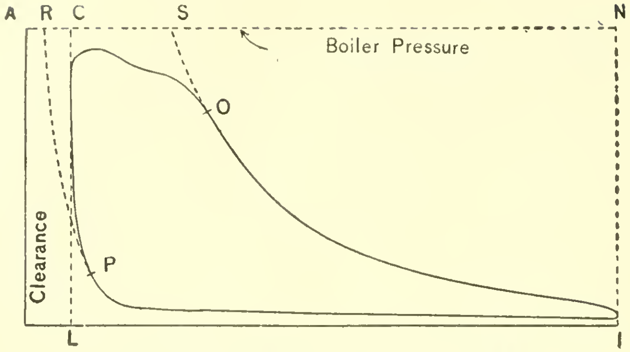

Fig. 23 Diagram Factor Net Volume

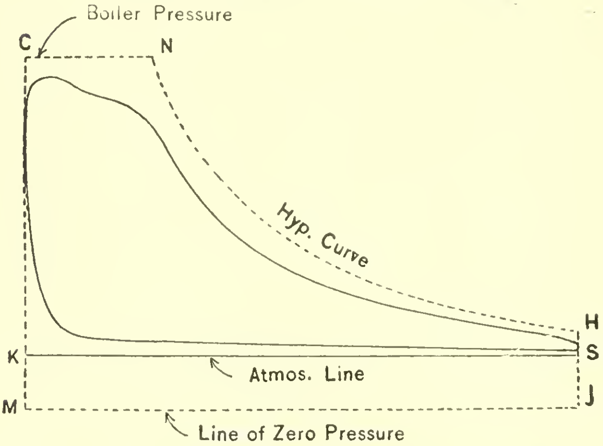

Fig. 24 Diagram Factor Non-Condensing Engine

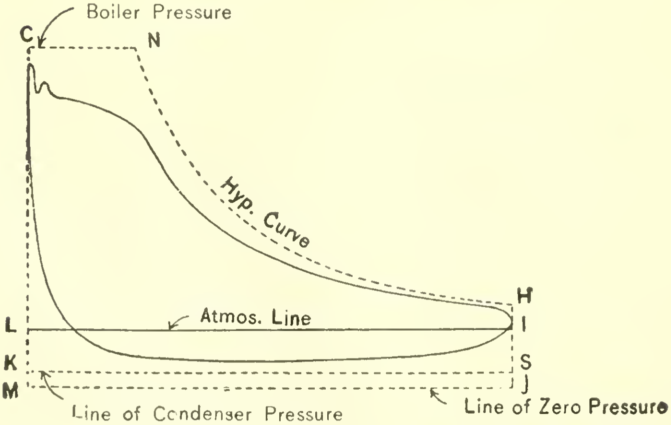

Fig. 25 Diagram Factor Condensing Engine

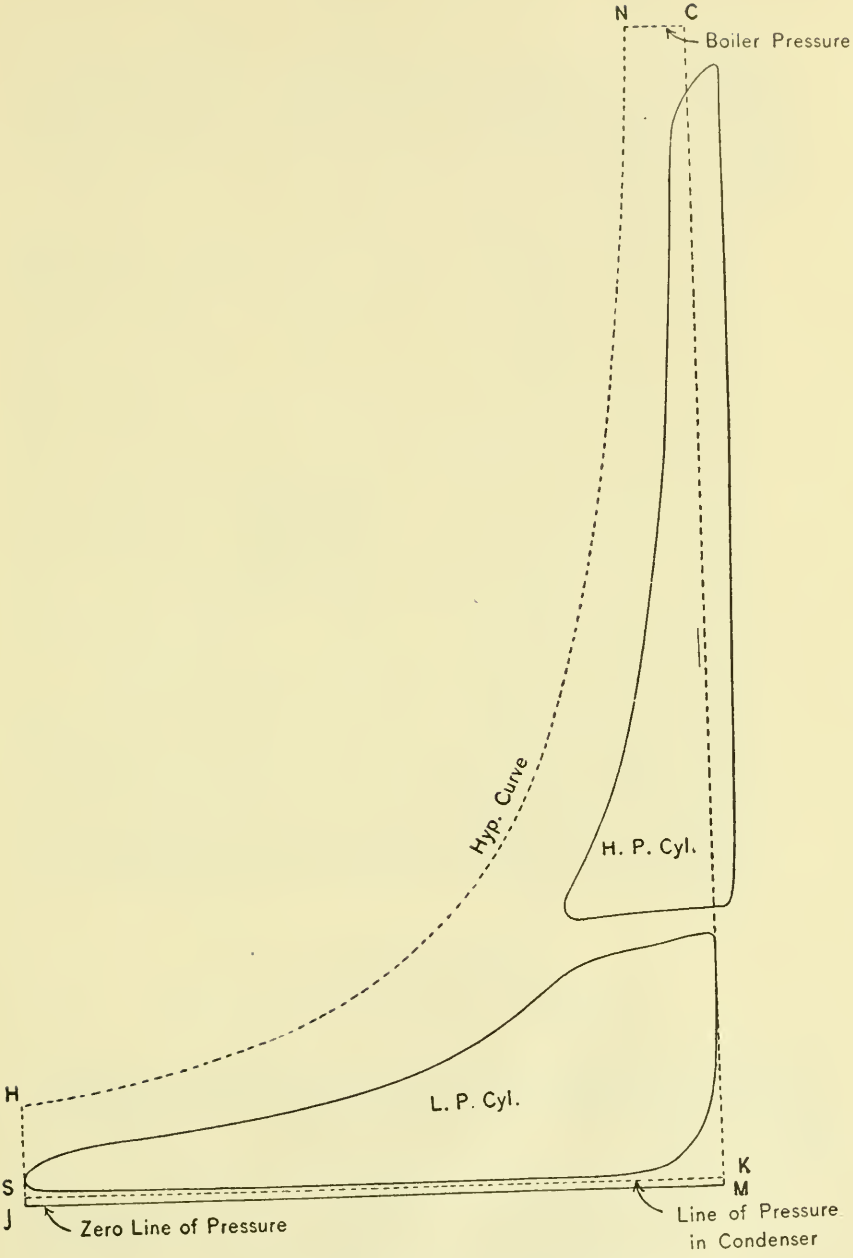

FIG. 26 DIAGRAM FACTOR COMPOUND ENGINE

APPENDIX NO. 29

HEAT ANALYSIS

Reference may be made to standard textbooks on thermodynamics for detailed information regarding the methods to be pursued and the formulæ to be used in working out the heat analysis in a steam engine required in research work. The principles involved may be briefly summarized as follows:

> A certain amount of heat is supplied to the engine in a given time, and this is represented by the number of lb. of steam supplied, multiplied by its total heat. A part of this heat is used in the jacket, if one be employed, and the balance passes through the cylinder. That entering the jacket is lost partly by radiation from the outside surface, and the remaining portion enters the walls of the cylinder and is taken up by the steam within it. The operations going on within the cylinder consist, first, of the transfer of a portion of the entering heat into a small portion of the thickness of the walls of the cylinder, heating them to the temperature of the entering steam. This transfer of heat is most active during the period of steam admission and up to the point of cut-off. After the cut-off, the transfer continues until the lower pressure due to expansion reaches such a point that the temperature of the steam is below that of the interior surfaces last uncovered. Then the interchange of heat is reversed, and the metal gives up heat to the steam and causes what is known as re-evaporation of the particles of water on the surface of the cast-iron walls and piston. The giving up of heat from the small thickness of the surfaces which were heated during admission begins after cut-off takes place, or, to be more explicit, after the pressure begins to fall, and continues during the entire stroke. Heat is also given off from the metal of the cylinder to the steam, due to the higher temperature of the jacket, if any be used, and this occurs throughout the whole of the stroke, assuming, what is usually true, that the pressure of steam in the jacket is greater than that in the cylinder.
>
> There is another action going on in the cylinder and that is the loss of heat due to the performance of work. There is also a loss of heat, due to radiation from those portions of the cylinder not protected by the jacket. The quantity of heat remaining after the steam has been subjected to these various complicated actions is that which passes out with the steam through the exhaust valve and is thereby "rejected," as it is termed, to the condenser or to the atmosphere. The heat analysis determines the amount of heat which is used and transferred in these various actions during the time in question, and the various percentages which these amounts bear to the total quantity of heat supplied.

APPENDIX NO. 30

TEMPERATURE-ENTROPY DIAGRAM

The study of the heat analysis is facilitated by the use of the temperature-entropy diagram, in which areas represent quantities of heat, the coordinates being the absolute temperature and entropy. Such a diagram is shown in Fig. 27*.

When the quantities given in the steam tables are plotted, two curves, *AA* and *BB*, are obtained, which may be termed the water line and the steam line, *AA* being the logarithmic curve if the specific heat of the water is taken as constant. The diagram refers to a unit weight of the agent. The heat necessary to raise a pound of water from the temperature *ma* to the temperature *pa'* and evaporate it at that temperature is represented by the area *aa'b'qm*. If the steam be now expanded adiabatically the temperature will fall to *qs* and x per cent $=\frac{as}{ab}$ will remain as steam, the rest being liquefied. If the steam is now rejected, it carries away with it the heat *sqma*, the work area being *a'b'sa*, from which must be deducted the work w (expressed in heat units) to pump a pound of water into the boiler The efficiency of this cycle is evidently

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* Reproduced from Trans. Am. Soc. M. E., vol. 24, p. 758, fig. 134.

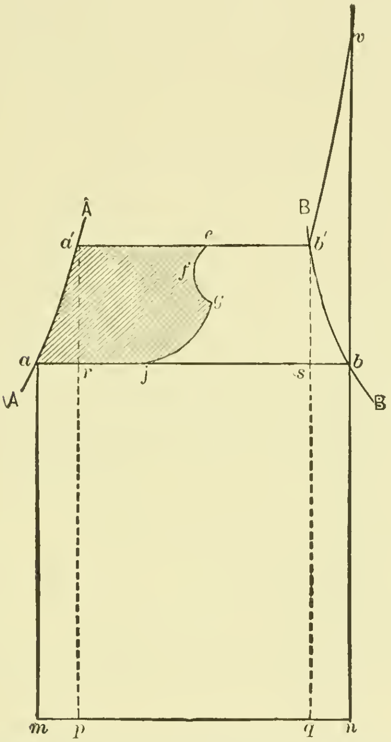

Fig. 27 Temperature-Entropy Diagram

$$\frac{h + L_1 - xL_2 - w}{h + L_1}$$

in which

$$x = \frac{ar + a'b'}{ab} = \log_e \frac{T_1}{T_2} + \frac{L_1}{T_1} + \frac{L_2}{T_2}$$

By the action of the walls a portion of the steam is liquefied prior to the expansion, which therefore begins at *c*, and since the cooling action of the walls continues, the expansion line falls off to *ef*, from which point a reverse action takes place, and the expansion line bends over to *g*. Finally, since the release takes place before the condenser temperature is reached, the heat rejection starts at *g*, following a line of equal volume until the exhaust port temperature is reached at *j*. If heat is added during expansion enough to keep the steam theoretically saturated, as, for example, by a water jacket, such additional heat is represented by the area *b'bnq*, and the additional work obtained by the triangle *b'bs*. If the steam is superheated sufficiently to give by expansion theoretically dry steam at the end,

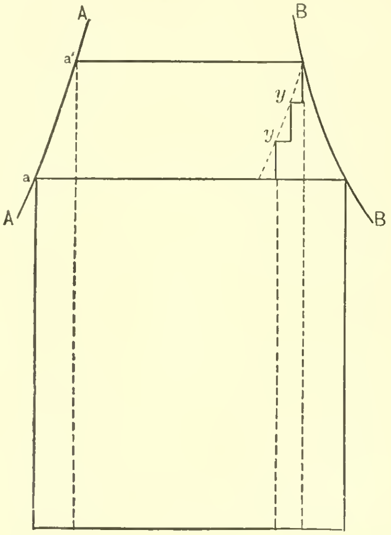

FIG. 28 TEMPERATURE-ENTROPY DIAGRAM

such additional heat is represented by the area *b'vnq* and the additional work by *b'vbs*. Neither of these extra amounts of work are realized in practice, and it is evident from the diagram that the heat thus applied is in both cases less efficient than in the principal cycle. Nevertheless the action in each case is to bring the point *c* nearer the point *b'*, and to effect a notable net economy.

The Carnot cycle would be obtained if in the Rankine cycle the rejection of heat were stopped at *r*, and the temperature of the mixture raised to *a'* by compression. This cannot be practically accomplished, but a system of feedwater heaters has been suggested and exemplified in the Nordberg engine, which is theoretically a close equivalent to it. Where steam is expanded in, say, three cylinders, the feedwater may be successively heated from the receiver intermediate between each pair, the effect of which is illustrated in Fig. 28*. The expansion line follows the heavy line, being carried over to

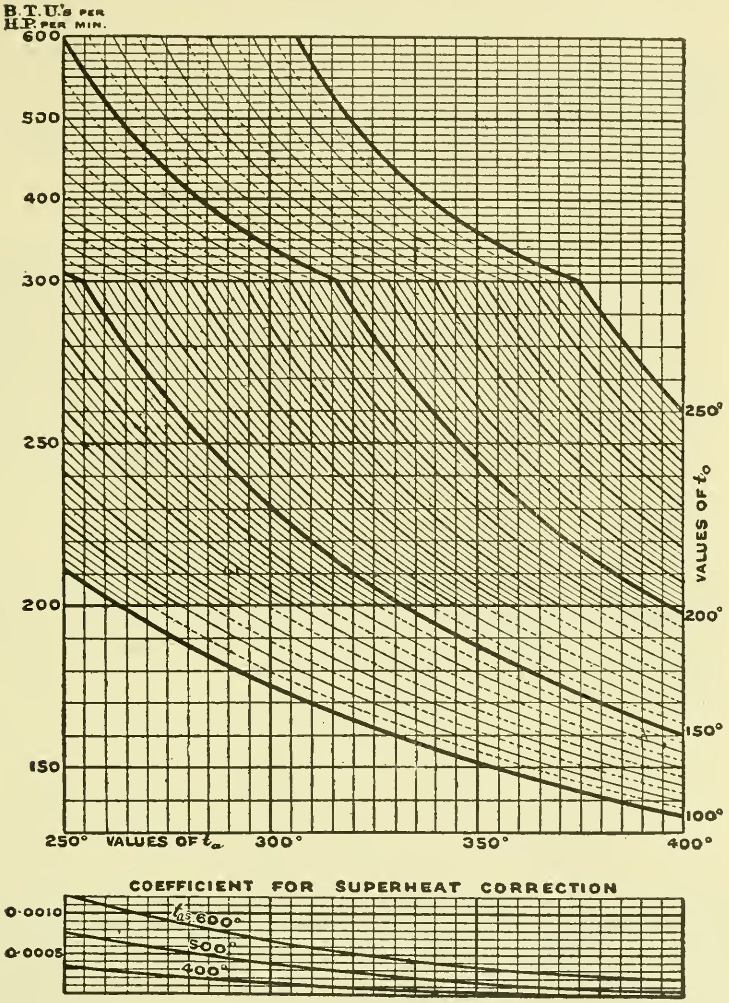

Fig. 29* Heat Units Expended per i.h.p. per Minute by Ideal Steam Engine Conforming to Rankine Cycle

y by the first feedwater heater and to *y'* by the second feedwater heater. With an infinite number of such feedwater heaters, the line *yy'* would be parallel to *aa'*, and the cycle equivalent to that of Carnot.

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* Reproduced from Trans. Am. Soc. M. E., Vol. 24, pp. 759, 761, figs. 135, 136.

APPENDIX NO. 31

RATIO OF ECONOMY OF AN ENGINE TO THAT OF AN IDEAL ENGINE

The ideal engine is assumed to be the same as that used for the British Institution of Civil Engineers Standard. It is one which conforms to the Rankine cycle, in which an ideal cylinder with no clearance is used. Steam is admitted at constant pressure and expanded adiabatically to the back pressure.

The ratio desired is obtained by dividing the heat consumption of the ideal engine by that of the actual engine.

To facilitate the determination of this ratio the heat consumption of the ideal engine may be obtained from the chart given in Fig. 29 which is copied from the Proceedings of the Institution of Civil Engineers. In this chart t_s represents the temperature of saturated steam at the boiler pressure in deg. fahr.; t_{sa} that of the steam furnished to the engine should there be superheating; t_e that of the exhaust. The heat units consumed per minute per i.h.p. can be read off directly from the curves given in the upper portion of the diagram; thus if the temperature of the exhaust (t_e) is 212 deg. fahr., and the temperature of the steam at boiler pressure is 350 deg. fahr., the heat consumption is 265 heat units per i.h.p. per minute. If the steam is superheated, the quantity is corrected by employing the factor obtained from the lower part of the diagram. Opposite the temperature of saturation corresponding to the pressure in the boiler, and on the curve corresponding to the temperature of superheated steam (t_{as}) is found a coefficient. This coefficient multiplied by the exhaust temperature, and by the heat consumption per minute, gives the deduction to be made on account of the superheating. Thus, if the temperature of the superheated steam is 500 deg. fahr. in the case noted, we find opposite 350 deg. for t_s, and on the curve for $t_{as} = 500$ deg., the coefficient 0.00015. The correction is $0.00015 \times 212 \times 265 = 8.5$ B.t.u., and the heat consumption with superheated steam is $265 - 8.5 = 256.5$ B.t.u. per i.h.p. per minute.

APPENDIX NO. 32

LEAKAGE TEST OF INSIDE PLUNGER PUMP

The leakage of an inside plunger is most satisfactorily determined by making the test with the cylinder head removed. A wide board or plank may be temporarily bolted to the lower part of the end of the cylinder, so as to hold back the water in the manner of a dam, and an opening made in the temporary head thus provided for the reception of an overflow pipe. The plunger is blocked at some intermediate point in the stroke (or, if this position is not practicable, at the end of the stroke), and the water from the force main is admitted at full pressure behind it. The leakage escapes through the overflow pipe, and it is collected in barrels and measured.

Should the escape of the water into the engine room be objectionable, a spout may be constructed to carry it out of the building. Where the leakage is too great to be readily measured in barrels, or where other objections arise, resort may be had to weir or orifice measurement, the weir or orifice taking the place of the overflow pipe in the wooden head. The test should be made, if possible, with the plunger in various positions.

In a case where it is difficult to remove the cylinder head, it may be desirable to take the leakage from one of the openings provided for the inspection of the suction valves, the head remaining in place.

It is here assumed that there is a practical absence of valve leakage, a condition of things which ought to be attained in all well-constructed pumps. Examination for such leakage should be made first of all, and if it occurs and it is found to be due to disordered valves, it should be remedied before making the plunger test. Leakage of the discharge valves will be shown by water passing down into the empty cylinder at either end when they are under pressure. Leakage of the suction valves will be shown by the disappearance of water which covers them.

If valve leakage is found which cannot be remedied, the quantity of water thus lost should also be tested. The determination of the quantity which leaks through the suction valves, where there is no gate in the suction pipe, must be made by indirect means. One method is to measure the amount of water required to maintain a certain pressure in the pump cylinder when this is introduced through a pipe temporarily erected, no water being allowed to enter through the discharge valves of the pump.

APPENDIX NO. 33

LOCOMOTIVE TESTING STATIONS

The shop test described in the Locomotive Code cannot readily be made unless a testing station is available where the locomotive may be sent, and where this work may be carried on. Such a station is too expensive to be constructed for the purpose of ascertaining the performance of an individual locomotive. How to arrange and install a shop-testing plant seems therefore a matter for independent consideration.

A number of universities have these plants in use or in process of installation, a notable instance being the Purdue University, which was the first to inaugurate the shop test.

The most elaborate work that has recently been done in the shop-testing line is that which was carried on in 1904 by the Pennsylvania Railroad Company at the Louisiana Purchase Exposition, St. Louis. A full description of the plant there installed, and of the tests made upon it, may be found in a volume published in 1905 by that company, entitled "Locomotive Tests and Exhibits." A study of this book is commended to any who require detailed information regarding the design, construction, and operation of shop-testing apparatus. The plant and tests referred to were planned under the counsel of an advisory board, three of whom were members of the Society.

APPENDIX NO. 34

LOCATION OF APPARATUS AND INSTRUMENTS FOR LOCOMOTIVE TESTS

The location of apparatus and instruments in a shop test do not need to conform to restrictions of space such as pertain to road tests, and substantially the same considerations apply in that case as in the ordinary stationary steam plant. Reference may therefore be made to the Boiler and Engine Codes for any needed instructions in this matter.

As to the locations and requirements for road tests, the following directions should be observed:

(1) The water meter should be attached to the suction pipe of the injector at a point where it can be conveniently read when the train is in motion. A check valve should be provided to prevent hot water backing through it when starting and stopping the injector, as also a strainer.

(2) The indicator driving rig should be some form of pantograph motion, with a light rod transmitting the reduced motion to a point near the indicator. Lack of room and facility of operation make it desirable to use a single indicator for each cylinder, connected to the two ends by a three-way cock. It is best to carry the pipes to the side of the cylinder rather than the heads, and provide a branch leading to the steam chest, sharp bends being avoided, and the outside protected from radiation. Absolute rigidity of the indicator cock is essential, and it should be obtained by clamping it securely to the cylinder.

(3) A special steam gage should be employed which can be read by the observer stationed in the pilot box at the front end.

(4) A rod should be attached to the reverse lever and carried forward to the pilot box, where a scale is provided to show its position. A rod should likewise be connected to the throttle-valve lever for the same purpose.

(5) The draft gage may be a simple U-tube containing water, and it can best be placed in the pilot box. A rubber tube connection should be provided which may be attached at will to pipes leading to the smoke box and furnace.

(6) The flue thermometer should be one graduated to 1000 deg. fahr., and inserted so that the bulb occupies a position below the tip of the exhaust nozzles and between them and the front tube plate of the boiler.

FOREIGN REVIEW

BRIEF ABSTRACTS OF CURRENT ARTICLES IN FOREIGN PERIODICALS

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FOREIGN REVIEW

Considerations of space make it impracticable to give equally full abstracts of all papers in foreign periodicals containing matter of interest to the American mechanical engineer. Not more than a few lines sometimes can be given to an article in the FOREIGN REVIEW. The editor would be glad, however, to receive inquiries for further information in connection with such short abstracts, and will so far as possible reply either by letter or by the publication of fuller abstracts in subsequent issues of THE JOURNAL. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *t* theoretical. Articles of exceptional merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

The abstracts in the Foreign Review, with few exceptions (as in the case of articles running through more than one issue), are made from publications which have reached the Library of the Engineering Societies during the preceding month. Even with a selection thus limited to one month one cannot help being impressed by the wide range of investigation covered by European laboratories and the readiness with which manufacturers on the other side of the water adopt the most modern methods.

THIS MONTH'S ARTICLES

Besides notices of several investigations made in Holland and Germany in connection with concrete and reinforced-concrete practice, there are given this month accounts of experiments with methods of treating boiler sheets to avoid initial stresses and cracking when assembling and riveting the boiler shell. Experiments upon gas velocity and energy of flow are also outlined.

The results of such investigations are sometimes negative in character, as when it is shown that sulphur in fuel oil used in an internal-combustion engine does not injure it; or that a cheaper process or material may be used just as well as one more expensive, cast-iron pipes in economizers being as good as or better than wrought-iron ones, or slag cement practically equal to portland cement. On the other hand, new apparatus sometimes results from such experiments. For example, W. Gaede, from what appeared a purely theoretical investigation of external friction between gas molecules and the walls of a vessel, and the angle of rebound of the molecules, designed a new vacuum pump, exceeding mercury vacuum pumps in speed and flexibility of operation. That manufacturers are not slow to take advantage of experiments, is shown in the installation for testing materials of the Protos Automobile Company of Nonnendamm, which includes apparatus for metallographic and microphotographic tests and the application of the latest processes such as the Heyn etching process, and which not many universities in this country or abroad can equal for completeness.

More and more space in engineering publications is being taken by new problems hardly mentioned a few years ago. Thus, whole issues last month were devoted to heavier-than-air flying machines by Der praktische Maschinen-Konstrukteur (no. 18) and Mémoires de la Société des Ingénieurs Civils de France (July), besides a large number of articles in other periodicals, one of which, by M. Rateau, from a paper read some time ago before the Association Technique Maritime, will be abstracted in the next number of The Journal. Some attempts at solving another new problem, driving an internal-combustion engine on stored oxygen under the peculiar working conditions existing in a submarine boat, are reported in this issue.

Among articles of a theoretical character, attention is called to the continuation of Karl Mayer's investigation on the theory of plunger pumps, and Professor Dwelshauvers-Dery's very favorable criticism of J. Paul Clayton's method of logarithmic indicator diagrams, presented at the Spring Meeting of the Society in Cleveland. Mr. Mayer is the author of an important work on the theory and design of plunger pumps and is considered an authority on this subject in Germany. His article, together with the Borsig tests of duplex-spring valves in pumps, of which an abstract was given in the October issue of The Journal, form a noteworthy contribution to the theory of this type of machinery.

Professor Dwelshauvers-Dery calls attention to the fact that the usual steam engine indicator diagram made up of the head and crank diagrams does not at all represent the average conditions in the engine cylinder, and that, moreover, the law of compression of steam in the engine cylinder differs from that of expansion. It is a rather interesting fact, considering that the engineering world has been using the steam indicator, in one form or another, almost ever since the days of James Watt, that Prof. Dery, writing in the *Revue de Mécanique*, a highly technical publication, still finds it necessary to emphasize the necessity of certain precautions in using this instrument.

Air Conditioning

METHODS OF VENTILATION AND REFRIGERATION OF ELECTRIC PLANTS (*Procédés de ventilation et de réfrigération des installations électriques*, H. Marchand. *L'Industrie électrique*, vol. 21, no. 497, p. 397, September 10, 1912. 3 pp. *p*). General discussion of ventilation and methods for keeping cool electric generating stations.

Air Machinery

STEEL PLANT BLOWING ENGINES BUILT BY LEFLAIVE & CIE (*Machines soufflantes d'aciéries construites par la maison Leflaive & Cie. Le Génie Civil*, vol. 61, no. 19, p. 373, September 7, 1912. 4 pp., 10 figs. *d*). Reciprocating steam engines with two parallel cylinders, according to the author, are better adapted for driving blowers in steel plants than either steam turbines or gas engines: the blowing of a converter is an operation of only a few minutes during which the engine has to reach from zero, its maximum speed, and return to rest; the air pressure varies from 0 to 3 kg, or 42.6 lb. per sq. in., and the power developed from zero to thousands of horsepower. The reciprocating steam engine can satisfy better than any other motor the required conditions of intermittent working, rapid starting and stopping, working at constantly varying speed and output. When the blowers are driven by gas engines, the problem is somewhat simplified by having the engine work all the time, though at various rates; but this is not economic.

The described blower is provided with Hoerbiger valves. The article contains a detailed description of the Collmann steam engine with the Collmann valve gear with catch.

Fuels

FABRICATION OF LEAN GAS BY DRY DISTILLATION OF SLUDGE FROM SEPTIC TANKS (*La fabrication de gaz pauvre par distillation pyrogénée des boues de fosses septiques*, L. Cavel. *Le Génie Civil*, vol. 61, no. 21, p. 418, September 21, 1912. 2 pp. *g*). Discussion of the question of disposal of the sludge from septic tanks first by drying it by exhaust steam from the pumping machinery, and then gasifying it for power purposes. *Sludge gas* is claimed to have a high calorific value. The article does not report any actual application of this scheme.

Briquetting of Coal with the Addition of Naphthaline (*Steinkohlenbrikettierung mit Naphtalinzusatz*, Grahn. *Glückauf*, vol. 48, no. 38, p. 1536, September 21, 1912. 1½ pp., 2 figs. *d*). The first process used in Germany of *coal briquetting with naphthaline as a binder* was that of Buss-Fohr in which naphthaline was introduced together with superheated steam having a temperature somewhat higher than the boiling point of the naphthaline, the idea being that the naphthaline would evaporate, and thoroughly permeate the coal mass. The temperature of the steam proved, however, to be too low, and no higher temperature could be used owing to the difficulty of handling in the presses masses that are too hot.

The Shüring process described in the present article avoids this difficulty first by evaporating the naphthaline in a special gasifier, and then introducing it by steam pressure into the deeper zones of the coal dust. Tar is still used as a binder, but instead of 6½ per cent required when no naphthaline is added, good briquettes can be produced with 4½ to 5 per cent tar and only 0.3 per cent naphthaline.

Measurement of Coal Consumption in Revolving Tubular Kilns by Flue Gas Analysis (*Messung des Kohlenverbrauches von Drehrohröfen durch Rauchgasanalyse. Tonindustrie-Zeitung*, vol. 36, no. 106, p. 1147, September 7, 1912. 3 pp. *pt*). Account of a paper presented by Dr. H. Kühl at the last annual meeting of the Society of German Portland Cement Manufacturers on the determination of *coal consumption of revolving tubular kilns.* The determination of coal consumption by weighing the coal and clinker is unreliable, because the fuel has to be measured before entering the furnace, while the clinker is measured after leaving the furnace and rotary coolers. There is therefore a considerable period between the two measurements, and to obtain any reliable results the duration of the test must be so long as to make the time difference between the measurements comparatively insignificant. Each test would then be conducted for several days, during which a perfect uniformity of operation must be maintained, a thing manifestly difficult and inconvenient. A better way of estimating the consumption of coal has been found in flue gas analysis.

The principle of the method consists in analyzing the mixtures of gases produced by the simultaneous process of combustion of a definite amount of coal and disoxidation of a certain amount of raw material, and calculating from this analysis the ratio of the two processes, i. e. how much coal was required for the disoxidation of a given amount of raw material, or burning of a given amount of clinker. The following formula is deduced:

$$x = \frac{K_1 V}{K_2 W - K_3 V}$$

where x is the number of *kg* of coal required to produce 100 kg of ash-containing clinker, such as leaves the kiln; $K_1 = \frac{400_m}{100-_n}$ where m is the contents in per cent of $CaCO_3$ in the raw material, and n loss due to burning; $K_2 = \frac{b}{3} + c + \frac{ad}{7} - \frac{e}{8} + \frac{f}{8}$, where b is the content of carbon in the coal, c that of hydrogen, d of nitrogen, e of oxygen, and f of sulphur, all in per cent; $\alpha = 0.262$, the volumetric ratio of oxygen to nitrogen in air; $K_3 = \frac{b}{3} - \frac{mg}{25\ (100-n)}$, where g is the content of ash in coal in per cent; and finally $V = \alpha s - q + \frac{t}{2}$, and $W = p + t$, where p,q,s,t represent the contents, in per cent, in the flue gases of carbon dioxide, oxygen, nitrogen, and carbon monoxide respectively.

The following example is used to show the simplicity of this method of determination. From the analysis of the raw material and coal the following values have been obtained for the constants: $K_1=5200$, $K_2=315$, $K_3=272$. Hence

$$x = \frac{5200V}{315W-272V}$$

From the gas analysis it was found in a special case that $p=27.6$, $q=0.2$, $t=0.2$, and $s=72.0$, all in per cent. Hence:

$$V=0.262\ .\ 72.0-0.2+0.1=18.8$$

$$W=27.6+0.2=27.8$$

Substituting these values in the formula for x, the following is obtained:

$$x = \frac{5200\ .\ 18.8}{315\ .\ 27.8-272\ .\ 18.8} = 26.8 \text{ kg per 100 kg clinker}$$

Since in regular kiln operation the contents of carbon monoxide in the flue gases are negligible, t may be taken as equal to zero. This not only considerably simplifies the formulae, but allows the use of the usual self-registering apparatus for the oxygen and carbon dioxide contents, which, together with the simple analysis of coal, afford an easy and convenient method of controlling the coal consumption of rotary kilns. A full derivation of the above formulae is given in the article.

Hydraulics

BLUE COAL (*La Houille bleue*, S. F. *La Lumière électrique*, vol. 19, no. 37, p. 340, September 14, 1912, 4 pp. *t*). The French call waterpower White Coal, and Jules Severin, in his paper before the Dijon meeting of the French Association for the Advancement of Sciences, called by analogy the *force of the tides* Blue Coal. The author proposes to construct a reservoir of about 1 km sq., and 7 to 14 m (23 to 46 ft.) deep, to take in water at high tide, and deliver it to water turbines later; the power is to be utilized according to the Taylor system used at the Rockland plant.

SHUTTING-OFF DEVICES ACTUATED BY THE EXPANSION OF LIQUIDS (*Absperrorgane durch den Auftrieb von Flüssigkeiten betätigt*, J. Fichtl, *Gesundheitsingenieur*, vol. 35, no. 36, p. 701, September 7, 1912. 6 pp., 13 figs. *dg*). General discussion of the principles of action of shutting-off devices, and description of some German types. The author investigates the mutual relations between the form of the float, its size and closing time and establishes, among other things, that the relation between the time and amount of liquid flowing out is represented not by a straight line, but by a logarithmic curve.

TO THE DETERMINATION OF PULSATIONS IN THE FOREBAY (*Zur Ermittlung der Schwingungen im Wasserschloss*, Ph. Forchheimer. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 32, August 10, 1912. 1½ pp. *t*). The author shows how the pulsations in a forebay may be determined without the usual limiting assumptions, such as the proportionality of pressure loss to velocity, or the use of small final differences instead of differentials. He deduces the following equation:

$$mZ+1-\ln\ (mZ+1)=mh+1$$

for the application of which the following data must be known: velocity of water in the state of resistance u_0, cross-section of penstock F, water area F_1, length of penstock l, loss of pressure head in the penstock $\frac{lu^2}{c^2R}$, then $m=2gF_1:c^2RF$; h is the loss of pressure head in the penstock in the state of resistance. A numerical example is given calculated by the author's formula and, for the sake of comparison, by the method of final differences.

Internal Combustion Engines

ELECTRIC POWER PLANTS IN MINE AND STEEL WORKS IN RHINELAND-WESTPHALIA, BELGIUM, NORTHERN FRANCE AND ENGLAND (*Elektrische Kraftanlagen in Rheinland-Westfalen, Belgien, Nord-Frankreich and England*, K. Hoefer. *Zeits. für des gesamte Turbinenwesen*, vol. 9, nos. 22 to 24, August 10, 20, and 30, 1912. 10 pp., 20 figs. The same article in a somewhat abridged form in *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 32, p. 1281, August 10, 1912. 10 pp., 25 figs. *d*). Description of *power transmission and prime movers* in the electric power plants in mine and steel works in the above named countries, with the description of some constructive details (mainly in *Zeits. für Turbinenwesen*). The article contains a table showing the *reserve power in power plants* in various countries. On the average about a third of the power of the machinery is held in reserve in case of accident, this reserve being practically equal for steam and *gas engines*, which shows that the latter are considered fully *reliable* in operation. The *smallest reserve* is kept in the case of *steam turbines*.

TESTS OF A JAHNS ANNULAR PRODUCER AT THE MINE HOHENZOLLERN (*Versuche an dem Ringgenerator-System Jahns auf der Hohenzollerngrube. Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 35, no. 37, p. 385, September 13, 1912. 3 pp., 5 figs. *de*). In the *Jahns producer* four chambers are built in a ring, with a common vertical passage along the axis. This passage is connected with the chambers above by openings provided with closing doors, and with those below by a circular duct. The purpose of combining the four chambers into a single ring is to conduct the gases (products of incomplete combustion), from a freshly charged chamber into the other chambers which have been in operation longer and so had time to become hot. The gases rich in coal-tar are forced, together with a steam jet, under the glowing fuel bed, and are thus transformed into permanent gases free from tar. The article contains a detailed description of operation and full data of tests, the gas produced being used for heating steam boilers.

Gas Engine-Dynamo Units (*Gasdynamos. Zeits. für Dampfkessel und Maschinenbetrieb*, vol 35, no. 38, p. 395, September 20, 1912. 3 pp., 6 figs. *dp*). Description of some German gas engine-dynamo direct-coupled units, with some suggestions as to the design of the gas engines in such units.

THE VALVELESS MOTOR AND GEAR OF THE FISCHER CAR, ZURICH (*Der ventillose Motor und das Getriebe des Fischer-Wagens, Zürich. Auto-Technik*, vol. 1, no. 17, appended to *Allgemeine Automobil-Technik*, vol. 3, no. 37, September 13, 1912. 3 pp., 8 figs. *d*). *The Fischer slide-valve motor* belongs to the same class as the Mustad motor described in THE JOURNAL, March 1912, p. 414, in that it has separate valves for the admission and exhaust, each valve having the width of about a quarter of the cylinder circumference, and both being exactly similar and therefore interchangeable. Only half of the piston surface is covered by the valves, the other half, or rather the other two opposite quarters, lying against the water-cooled cylinder walls. The piston is provided with three piston rings which by their spring action help to obtain a tight fit between the valves and the outer wall; a fourth, stationary ring in the cylinder casing provides packing for the part where the piston does not reach. The article describes also the method of machining the valves, and the *change speed gear* of the car, which, according to the author, is suitable only for small and cheap cars.

GAS PRODUCERS FOR THE GASIFICATION OF LIQUID FUEL (*Gaserzeuger für Vergasung flüssiger Brennstoffe*, Gwosdz, *Braunkohle*, vol. 11, no. 23, p. 357, September 6, 1912. 10 pp., 11 figs. *d*). In the Rincker and Wolter process used in Watergrasmeer, near Amsterdam, Holland, for producing gas from tar, the liquid fuel is gasified by passing it through a layer of glowing coke, without the admission of steam. The producer consists of two shafts connected in parallel. The coke is heated to glowing first in one shaft, into which tar is then introduced to be gasified. It is kept there until the temperature reaches a certain limit when the gases are led out to be used. The connections are changed in such a way that the air blast, in passing through the first shaft, drives the gases, only partly decomposed owing to the low temperature, into the second shaft where they pass through the glowing fuel from top downwards and are converted into stable gases.

This process has, however, the following disadvantages: (*a*) the air blast is passed through one of the shafts twice, and in the meanwhile the shaft has time to cool down thus retarding the decomposition of the oil vapors; (*b*) during the blowing of the producer the gases escape and the oil vapor contaminates the atmosphere.

In the improved type of the Berlin-Anhalt Machine Construction Company both shafts are connected in series, and get the hot blast alternately, the one which the hot air current strikes first being connected with the apparatus using the gas. The blowing of the producer shafts in series permits of keeping both generators hot; while alternating the direction of blowing has the advantage of the shaft which is blown first being hot from the previous blowing. Both generators are therefore better heated and the decomposition of oil vapors proceeds more effectively. The article describes also the Pintsch, Blaugas, and several American types of producers for liquid fuel (E. C. Jones, Frost, Ammet, Hydrocarbon Converter Company).

Kopenhagen Fishing Exhibition of 1912 (*Kopenhagener Fischerei-Ausstellung 1912,* Gbn. *Die Gasmotorentechnik,* vol. 12, no. 6, p. 100, September 1912. 2 pp., 4 figs. *d*). Short descriptions of some *Scandinavian motors.* The motor of the *Bergsunds* Mek. Werkstads Aktiebolag in Stockholm has the *gasifier* as shown in Fig. 1, with a number of consecutively arranged narrow metal strips *H* connected with each other and with the lower side of the cover *I*. The fuel is thrown against these strips from the nozzle *G* as shown by the dotted lines, so that each strip receives and distributes in various directions an equal amount of fuel per unit of its area, this fuel being at the same time evaporated owing to the high temperature of the metal strips. This happens just before the piston reaches its highest position, the heat of compression together with that of the metal strips igniting the fuel mixture immediately after its admission. This effective system of gasification permits of admitting the fuel into the combustion chamber so late, i. e. so near the uppermost position of the piston, that all possibility of pre-ignition is eliminated, while the strips *H* are practically at constant temperature, independent of the load, owing to the cooling produced by the fuel being injected against them. The article also describes Tuxham and Junkers motors. (For other motors at this Exhibition, see *The Gas Engine,* vol. 14, no. 10, 1912, p. 541).

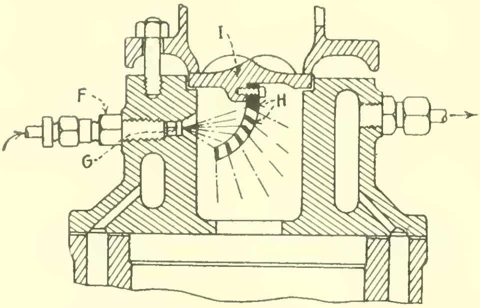

Fig. 1 Gasifier of the Bergsunds Motor

Gas Velocity and Energy of Flow (*Gasgeschwindigkeit und Strömungsenergie,* v. Löw, *Zeits. des Mitteleuropäischen Motorwagen-Vereins,* vol. 11, no. 17, p. 373, Mid-September 1912. 2 pp., 3 figs. *cp*). There are as yet no generally accepted rules for the *proportions of automobile valves and gas velocities.* Adler makes the suction and exhaust valve cross-sections equal, and uses 102 m (332 ft.) per sec. as the velocity of flow when the maximum velocity of the car is 125 km (77 miles) per hour.

Benz has the cross-section of the suction valve nearly twice as large as that of the exhaust valve, and the velocity of flow 45 m (147 ft.) per sec. for the maximum velocity of the car 134 km (82 miles) per hour. Horch uses a ratio of suction valve and exhaust valve cross-sections as 4 to 3, and for the inlet velocity of the charge 57 m (187 ft.) per sec. at the car speed of 87 km (54 miles). Although the most advantageous ratio of valve cross-sections can be found only by suitable tests, the author tries to establish certain facts in this connection by means of the diagram in Fig. 2 A., where *aa* is the atmospheric line, *vv* absolute zero or vacuum line, and the pressure process during the greatest part of the exhaust stroke is represented by the line 1-2. At the point 2 the piston is supposed to have reached the end of the exhaust stroke and to be in the neutral position, the pressure being not atmospheric, but above it, 2-3, owing to throttling in the exhaust outlet. When the piston starts on a new stroke, there is an expansion, but the atmospheric pressure is reached only at 4; it is therefore only from there on that the real suction of the piston begins, 3-4 representing the volumetric cylinder loss at the beginning of the suction stroke. This cannot be helped by opening the suction valve when the piston is in its neutral position because then the expansion in the suction piping forces back the fresh mixture which in that case does not begin to reach the cylinder until the piston is at 4. The throttling pressure 2-3 must therefore produce always the volumetric cylinder loss 3-4.

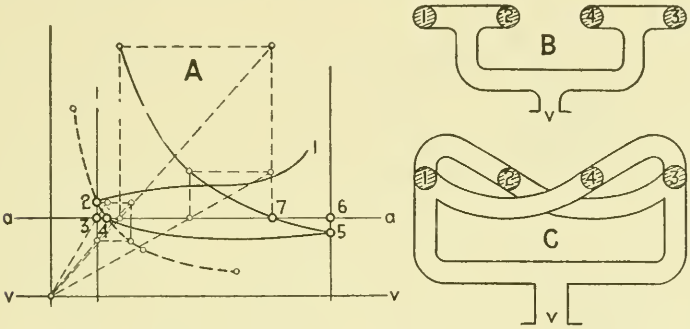

Fig. 2(A) Pressure Diagram for the Exhaust and Suction Stroke of an Automobile Engine (B) and (C) Suction Piping Connections in a Multi-cylinder Automobile Engine

Should the throttling at the end of the suction stroke be equal to that at the end of the exhaust stroke, or 5-6 be equal to 2-3, then, since the compression curve rises much slower than the expansion curve falls from 2 on, the volumetric loss 6-7 would be about six times as large as that at the beginning of the suction stroke, or 3-4, which shows that with equal throttling areas at the suction and exhaust valves the volumetric cylinder loss produced by the throttling at the suction valve is considerably greater than that produced by the throttling at the exhaust valve. The suction valve ought therefore to be larger than the exhaust valve.

It is, however, worth noticing that the Daimler Motor Company who were among the leading builders of engines with unequal valves, have now equal valves in their aeronautic motors.

It is usually accepted that the volumetric cylinder loss due to throttling at the valves may be reduced by late closing of the valves. Leading constructors are, however, by no means agreed as to just how late the closing should be, and practice varies between 0 and 49 deg. for the suction valve and 0 and 22 deg. for the exhaust valve. In multi-cylinder engines, owing to the energy of flow due to high gas velocities, it may be even advisable to select different closing times for the valves of different cylinders. Cylinder 1 in Fig. 2B must set in motion the gas in the left part of the suction piping, while cylinder 2 finds it already in motion, and therefore receives more gas with the same valve gear. The same happens with the cylinders 3 and 4 for the right side of the suction piping. A more uniform system of feed is represented when the suction piping is arranged as shown in Fig 2C: when there is suction in cylinder 1, it receives the larger part of its gas through the left side of the piping from the carbureter *c*; but there is an additional stream of gas coming past 4, after the closing of its valve after the completion of its previous suction stroke. When the suction stroke in 1 is ended, the stream of gas continues to flow first to 2, and then to 3 the last of which also receives some mixture direct from the carbureter, through the right side of the piping, this stream continuing to 4 after the end of the suction stroke in 3. The flow is therefore continuous and uniform, and there are no particular losses of the energy of flow. The author insists, however, on the necessity of a systematic series of experiments for the full elucidation of all these points.

On the Influence of Sulphur in Liquid Fuels used in Motors (*Uber den Einfluss des Schwefels in flüssigen Brennstoffen beim Motorenbetrieb*, E. Gräfe. *Der Ölmotor*, no. 2, 1912, through *Braunkohle*, vol. 11, no. 24, p. 383, September 13, 1912. *p*). In Diesel and other internal-combustion engines oils, mostly by-products of the coal-tar industry, are used containing up to 2 per cent of *sulphur*, and the question arises in how far this may *harm the engine*. The products of oxidation of sulphur, viz. sulphur dioxide and sulphur trioxide, do not attack iron unless they are dissolved in water, and form sulphurous or sulphuric acid. In the motor cylinder or in a boiler furnace where water is present only in the form of steam, there can be no corrosion due to the presence of sulphur, but wherever there is condensation of water, as in the exhaust piping, the iron is strongly attacked. This may be prevented, however, either by making the exhaust piping short, or by covering it with lead. On the whole, for the motor itself, the presence of sulphur is therefore practically harmless. Cp. *Fuel Oil Journal*, October 1912. p. 15.

New Working Processes for Internal-Combustion Engines in Submarines (*Neue Arbeitsverfahren für Verbrennungskraftmaschinen in Unterseebooten*, Dr. G. *Petroleum*, vol. 7, no. 22, p. 1249, August 21, 1912. 2 pp., 3 figs. *d*). Description of some new *submarine engines* and discussion of their *working processes*. Submarine engines during the run under water are sometimes fed by pure oxygen, the exhaust gases being used over again either in their original state, or without their carbon dioxide, i. e. with practically only their nitrogen. Contrary to this practice, Dr. Jaubert

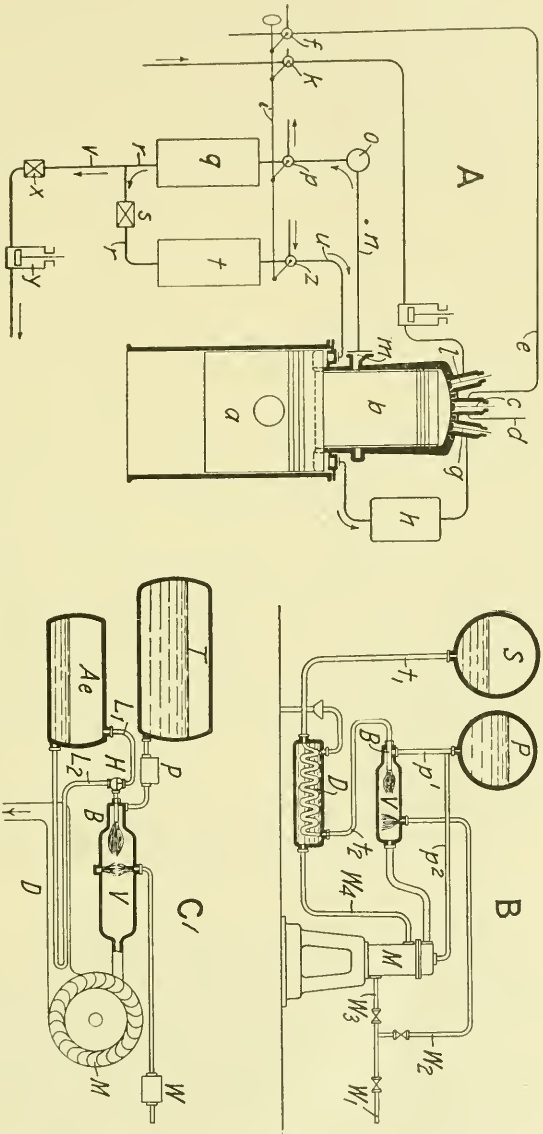

FIG. 3 SUBMARINE ENGINES WORKING ON STORED OXYGEN OF GASES RICH IN OXYGEN

of Paris recommends the elimination of all nitrogen from the exhaust gases and the use of only carbon dioxide, on the assumption that concentrated carbon dioxide in the absence of water vapor will take part in combustion in accordance with the following formula:

$$C_7H_{16}+4CO_2+\text{Heat}=5\tfrac{1}{2}C_2+8H_2O+\text{Heat}\ldots\ldots\ldots\ldots[1]$$

with the final products (coal and water) such that they can be conveniently thrown overboard. The engine is represented schematically in Fig. 3A. The pistons of the two cylinders which lie one above the other have a common connecting rod. Cylinder *a* acts as a compression pump and delivers the gas to *h*; the working cylinder *b* is provided with the valve *c* for the admission of petroleum coming through the pipe *d*, and injected under gas pressure (air or carbon dioxide), the compressed gas being brought to it by the pipe *e* provided with a three-way cock *f*. The pump *j* driven by the engine may be set in connection, by means of the three-way cock *k*, either with the atmosphere or the oxygen tank. The exhaust pipe *m* of the cylinder *b* communicates by the pipe *n* with collector tank *o* which in its turn may be connected by means of the three-way cock *p*, either with the atmosphere or with the compressor and separator *g*, the latter opening through the pipe *r* and pressure regulator *s* into the cleaner *t*. This cleaner is filled with peat or a similar mass, and by the three-way cock *z* may be set in communication either with the suction opening of the cylinder *a*, through pipe *u*, or with the atmosphere. The cocks *z,p,k,f* can be simultaneously regulated by the rod *i*. During the run above the surface the cocks *k,p*, and *z* are connected with the atmosphere, and the petroleum injection cock *f* with the compressed air tank. The piston *a* takes in air and delivers it to *h* whence it is taken for scavenging the cylinder *b*, while the pump *j* supplies atmospheric air for the explosive mixture to the same cylinder. During the underwater run the petroleum is injected by compressed carbon dioxide coming through the pipe *e*, the same gas being used also for scavenging. The exhaust gases are taken to the compressor and separator *g* in which the water vapor is condensed, and coal particles and unconsumed oil retained, while the clean carbon dioxide is sent on to the pipe *u* and cylinder *a*, and is later used for scavenging, as well as for the injection of fuel. Oxygen is introduced into cylinder *b* by the pipe *l*. Since no external air is introduced, the residual nitrogen is gradually expelled, and the exhaust gases contain practically only carbon dioxide and some oxygen. Simple combustion follows the equation

$$C_7H_{16}+11\ O_2=7\ CO_2+8\ H_2O\ldots\ldots\ldots\ldots[2]$$

and this, together with formula [1] gives the equation for the total reaction

$$2\ C_7H_{16}+11\ O_2+4\ CO_2=7\ CO_2+5\tfrac{1}{2}\ C_2+16\ H_2O\ldots\ldots\ldots\ldots[3]$$

In this process the residues of combustion contain a large amount of carbon dioxide thus considerably reducing the amount of this gas which would otherwise have to be stored on board the vessel, and only half the oxygen required according to equation [2] is used when the process thus takes place in accordance with equation [3]. A further advantage consists in the exhaust being only carbon dioxide soluble in water, and therefore attracting no attention of the enemy, as do air bubbles.

Another arrangement of submarine engines is shown in Fig. 3B, in which *nitrogen tetroxide* is used as the oxygen carrier. Since, however, both petroleum and nitrogen tetroxide are liquids, one of them must be evaporated before reaching the cylinder in order to produce a good atomization of the mixture. Nitrogen tetroxide, and not petroleum, is evaporated because it possesses a lower boiling point, so that cooling water which leaves the cylinder jacket at about 50 deg. cent. (122 deg. fahr.) can be used for this purpose (exhaust gases are not used since the necessary apparatus would take more room than can be given to it on board a submarine). *P* is the petroleum tank, *S* contains nitrogen tetroxide. The combustion chamber F is provided with the burner *B* which receives a small amount of petroleum through the pipe p_1, and vaporized nitrogen tetroxide throughout the pipe t_2, there occurring a partial combustion and decomposition of the nitrogen tetroxide which results in the production of a gaseous mixture rich in oxygen, while water is injected into the combustion chamber through the pipes W_1 and W_2, to reduce the temperature of the oxygen-carrying mixture. The comparatively cool mixture is then introduced into the cylinder of the motor *M* where it mixes with the petroleum brought by the pipe p_2. Pipes W_1 and W_3 bring the cooling water which after passing through the cylinder jacket continues to flow through the pipe W_4 to the vaporizer *D* where the nitrogen tetroxide is evaporated.

In another type the same inventor evaporates the fuel (benzine, benzole, etc.) by the heat of exhaust gases and uses as a source of oxygen *tetranitromethan* contained at atmospheric pressure in the tank *T*, Fig. 3C and brought by the pump *P* to the burner *B*. Gaseous ethane which has a pressure of 46 atmospheres even at the temperature of 4 deg. cent. (39.2 deg. fahr.), is contained in the tank *Ac*, and is brought to the burner by the three-way cock *H* and pipe L_1. The gases of combustion in the chamber F are so much cooled by the injection of water by pump *W* that they may be used to drive the *turbine M* from which the exhaust passes through the vaporizer *D* through which also passes the pipe L_2 from the ethane tank; after the engine started, the cock *H* is set so as to connect *Ac* and *B* by the pipe L_2.

Machine Shop

Spring Hammer (*Marteau à ressorts. Portefeuille économique des machines*, ser. 5, vol. 10, no. 681, p. 131, September 1912. 2 pp., 5 figs. *d*). Description of the Ajax spring hammer. Cp. The Journal, January 1912, p. 123.

New Dust Suction Device for Polishing Machines (*Neue Staubsaugevorrichtung für Schleifmaschinen. Giesserei-Zeitung*, vol. 9, 17, p. 548, September 1, 1912. 1 p., 2 figs. *d*). The designer's problem was to construct a dust suction apparatus which might be used in connection with polishing machines in which the polishing wheel does not remain in one position, but travels around the piece of work. Such a device, patented in Germany by the firm Frederich Schmaltz, in Offenbach-A-M., is schematically shown in Fig. 4. The suction device travels with the wheel around the work. The dust-catcher *b* is then in constant position as regards the polishing wheel *a*, while the suction piping *c*, attached to the dust catcher passes through the filter *d* and through the center of the rotary motion of the tool holder *e*, and finally to the bore in the hollow shaft *f*. The suitably shaped pipe *c* is fixed on the adjustable bracket *g* and therefore follows the motions of the tool holder. The other end of the hollow shaft of the polishing machine is air tight and rotatively connected with the suction piping *h*. Cp. description of a similar British device in Dust Collecting Device for Grinding Machines, *The Engineer* (London), September 13, 1912, p. 287.

Mechanics and Machine Design

Bending of Curved Tubes (*Die Biegung krummer Rohre*, H. Lorenz, *Dinglers polytechnisches Journal*, vol. 327, no. 37, p. 577, September 14, 1912. 4½ pp., 5 figs. *mt*). The author derives a new simple formula for the *variation of the curvature of a pipe* through an external moment, from the conditions of equilibrium of an element of the wall of a curved tube under the assumption of conservation of double symmetry of the tube cross-section. He investigates also the direction of the moment producing increase or decrease of the radius of curvature of the tube, and derives a simple formula for the total contraction of the circular cross-section of the tube in the radial or axial direction. The problem is treated by purely mathematical methods.

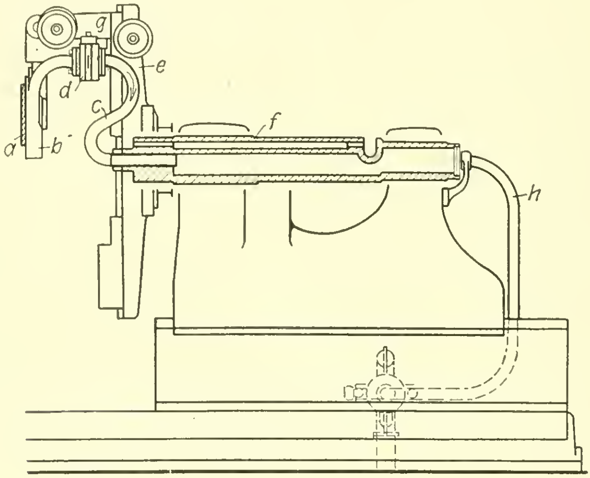

Fig. 4 Dust Suction Device for Polishing Machines

On Lubrication of Machine Parts (*Uber die Schmierung von Maschinenteilen*, L. Singer, *Petroleum*, vol. 7, no. 23, p. 1307, September 4, 1912. 12 pp., 64 figs. *d*). Discussion of the *theory of lubrication* mainly on the basis of Ubbelohde articles (see The Journal, June and August 1912, pp. 963 and 1245), and of the *design of machine parts subject to friction*, e.g. journals, bearings, etc., so as to obtain the best lubricating efficiency, as well as the apparatus for combined oil-and-graphite lubrication.

Experiments on the Determination of the Torsion of Bars with Rectangular Cross-Section, and Elongation and Contraction of Area of Bars under Tension (*Versuche über die Verdrehung von Stäben mit rechteckigem Querschnitt und zur Ermittlung der Längs- und Querdehnung auf Zug beanspruchter Stäbe*, O. Bretschneider. *Mitteilungen über Forschungsarbeiten*, no. 21, p. 1. 1912. 37 pp., 17 figs. *e*). Full reports of the experiments reported in The Journal, May 1912, p. 805, from a preliminary publication.

Staging Ratio in Speed-Changing Gears (*Abstufungsverhältnisse von Wechselgetrieben*, Fr. v. Löw. *Zeits. des Mitteleuropäischen Motorwagen-Vereins*, vol. 11, no. 16, p. 349, End August 1912. 4 pp., 4 figs. *et*). Discussion of the problem of staging ratios, that is, whether the first transmission should be equal exactly to a quarter of the transmission when directly coupled, or more or less, and how the intervals between single stages should be proportioned, these being, according to the author, questions on which constructors are not yet agreed. In the modern cars (only German types are considered) with three-stage speed changing gears the author found only one car with the ratio between the highest and lowest transmission greater than 4, while in nearly all cars with four-stage speed changing gears this ratio was more than 4. The lowest transmission was for hill climbing, the highest for level road work. Different demands should be made in these respects regarding three and four-stage cars, particularly as the high-power car is better able to take care of variable road conditions. According to the author there is no reason why three-stage speed changing gears should be used any more since the disappearance of the belt transmission has left their construction without any real justification. The second part of the article gives data from the author's experience with the speed-changing gears of German cars under road conditions, and discussion regarding them.

Resonance Phenomena in Suction Piping of Compressors and Gas Engines (*Phénomènes de résonance dans les conduites d'aspiration des compresseurs et des moteurs à gaz*, P. Voissel. Revue de mécanique, vol. 31, no. 2, p. 155, August 31, 1912. 7 pp., 11 figs. *et*). French translation of the article in the *Zeits. des Vereines deutscher Ingenieure* fully reported in The Journal, August 1912, p. 1248.

A New Axial Compression Formula (*En ny Knackningsformel*, Torbjørn Strand. *Teknisk Ukeblad*, vol. 59, no. 36, p. 460, September 6, 1912. 2 pp., 3 figs. *t*). Starting from the Schwartz-Rankine and Euler formulae for axial stresses in columns, the author by a rather complicated mathematical process derives the following *formula for axial stress*:

$$\beta_K = \frac{K}{1+\alpha\left(\frac{l}{i}\right)^2+\beta\left(\frac{l}{i}\right)^4+\gamma\left(\frac{l}{i}\right)^6}$$

where K is the compression strength of the material, $\frac{l}{i}$ slenderness ratio, α, β, γ, are defined by equations of their own, but are for each given material, functions of compression strength and modulus of elasticity.

The article also contains a table showing the relation between the slenderness ratio and axial stress.

Shock Absorbers and Elastic Shackles (*Les Amortisseurs et jumelles élastiques. La pratique automobile*, vol. 8, no. 173, p. 3683, September 10, 1912. 2 pp., 12 figs. *d*). Description of the main French types of shock absorbers and elastic shackles.

Reduction Gear for Coaxial Shafts (*Reduziergetriebe für gleichachsige Wellen*, R. Kommer.. *Motorwagen*, vol. 15, no. 26, p. 618, September 20, 1912, 3 pp., 5 figs. *d*). Description of the *Paul H. Peters epicyclic gear* consisting of two planet wheels each of which is free to make eccentric motions without changing thereby the mutual position of the teeth of the two wheels, one of these wheels engaging a fixed toothwheel coaxial with the driving shaft, and the other a wheel sitting on the driven shaft. With such an arrangement the gear ratio is determined by two members of which each may be either negative or positive, according to the ratio of the teeth numbers in the separate pairs of wheels, so that the sum of the two may be equal to zero without the number of teeth of the engaging wheels being equal to each other. Ratios of transmission from 2:1 to 20000:1 and more may be obtained, either in the same direction as the driving shaft or opposite to it, without varying the direction of the driving force. The driving and the driven axes are coaxial. At least a third of all the teeth are always in engagement, this permitting the use of small teeth and a small wheel even for the transmission of large forces. It is claimed that even with the highest ratios of transmission the friction loss is below 6 per cent. Fig. 5 represents a type in which the driven shaft 40 may be driven in any direction or stopped altogether. On the middle part 34 of the driving shaft 35 sits loosely the wheel body 36 provided with three different rims 37, 38 and 43* with internal or external toothing. In engagement with the wheel rim 38 is the toothed wheel 39 sitting on the shaft 40, the latter concentrically embracing the end journal of shaft 34. The second rim 37 is in engagement

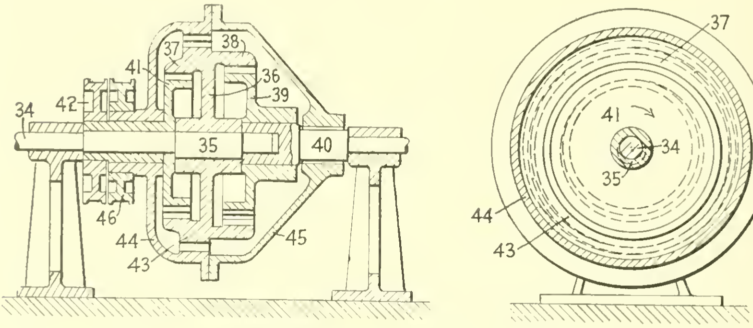

Fig. 5 Peters Epicyclic Reduction Gear

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* The original has 41, but that is evidently a mistake.—Editor.

with the toothed wheel 41 of which the long hub sleeve sits loosely on the shaft 34 provided with a brake pulley 42. To the third rim wheel 43 of the wheel body 36 corresponds the internal toothing of the casing member 44 which with its sleeve sits loosely on the sleeve of the toothed wheel 41 while another casing member 45, bolted together with 44, loosely encloses by its sleeve the shaft 40. The sleeve of the casing member is also provided with a friction pulley placed near that of 42. Suppose now that the brake pulley 42 is held fixed, and shaft 34 rotated in the direction of the arrow. Then the internal toothed rim 37 of the wheel body 36 sitting loosely on the eccentric journal 35, rolls along the wheel 41 held by the brake; the motion of the wheel 36 retarded thereby is transmitted through its toothed rim 38 to the toothed rim of the wheel 39, the sleeve of which is wedged fast to the shaft 40. The wheels 43 and 44 rotate in the same sense as the driving shaft. If on the other hand it is the pulley 46 that is braked, the rotation of the shaft 34 in the sense of the arrow produces an engagement of 43 with the now stationary internal toothing of the casing member 44, in a sense contrary to the direction of the arrow, the rotation of the wheel body 36 being as before transmitted to the shaft 40 through the toothed wheels 38 and 39, but in a direction opposite to that of 34. When both pulleys, 42 and 46, are loose, the whole gear rotates on 39, and with the slightest load on it the shaft 40 remains stationary. All the toothed wheels are in engagement all the time, but those which are not required to do any work are allowed to run loose.

The rest of the article contains rules for calculating the number of teeth in the various wheels, and a description of another type of gear based on the same principle, but said to be specially adapted for use in turbines, automobiles, and motor boats.

Pumps

Contributions to the Theory of Plunger Pumps (*Beiträge zur Theorie der Kolbenpumpen*, Karl Mayer, *Die Fördertechnik*, vol. 5, no. 9, p. 202, September 1912. 4 pp., 3 figs. *mt*). Continuation of the article abstracted in The Journal, October 1912, p. 1574, containing an investigation of the case when the *angle of coupling of a plunger pump is less than 90 deg.* Since the method of the author has been made clear in the preceding abstract, only his conclusions are reported here. By means of the combined process the coefficients of cyclic variation may be made equal to each other in six out of the eight intervals occurring in each revolution, and the maximum velocities in three out of four phases per revolution. The improvement, in per cent, in the coefficient of cyclic variation due to the use of the combined process (18.16) is greater than when the method of the most suitable angle of coupling (15.67) or that of the most advantageous ratio of cross-sections (10.52) are applied singly. In addition to that the combined process permits of freely choosing the ratio of cross-sections.

Each way of selecting the angle of coupling, i.e. greater or less than 90 deg., has its advantages. With the angle of coupling greater than 90 deg. the maxima may be obtained in all four phases, and acceleration pressure may be made equal in six out of the eight intervals per revolution, but the same value may be obtained for only four out of the eight intervals of coefficients of cyclic variations occurring per revolution. Six out of those eight intervals may be made equal, however, with the angle of coupling less than 90 deg., while equal maxima of velocity may be obtained only in three phases, and only four equal maxima of acceleration resistance. The angle of coupling has therefore to be chosen separately for each particular case in accordance with the purpose of the pumping installation. The article is to be continued.

External Friction of Gases and a New Principle for Air Pumps: Molecular Air Pump (*Die äussere Reibung der Gase und ein neues Prinzip für Luftpumpen: Die Molekularluftpumpe,* W. Gaede. *Physikalische Zeits.*, vol. 13, no. 18, p. 864, September 15, 1912. 6 pp., 5 figs. *d*). Description of a new pump based on the principle of utilizing the external friction between the gas molecules and the walls of a vessel, and the rebound of the molecules at a certain definite angle. In rapidity of operation and degree of exhaust the new pump exceeds the mercury vacuum pump. It can exhaust not only air, but also vapors.

Steam Engineering

On the Cycle of a Steam Engine (*Sur le cycle de la machine à vapeur*, A. Leduc. *La Revue électrique*, vol. 18, no. 209, p. 195, September 13, 1912. 3 pp., 3 figs. *t*). From a paper read by the author before the Nimes Congress of the French Association for the Advancement of Science. The efficiencies of the Carnot and Rankine cycles are supposed to be very close to each other; in a non-condensing engine working between 200 and 100 deg. cent. (392 and 212 deg. fahr.) they are, but in condensing engines working between 160 and 40 deg. cent. (320 and 104 deg. fahr.) the two efficiencies differ by 13 per cent.

The author proceeds to consider a theoretical case of engine with condenser working at 60 deg. cent (140 deg. fahr.), and using first saturated steam at 200 deg. cent. (392 deg. fahr.), and then steam produced in a boiler at 100 deg. cent. (212 deg. fahr.) and superheated to 200 deg. cent. Let T_0 and T_1 be the temperatures of the isotherms, k real specific heat of water at T_0, L_0 and L_1 latent heats of evaporation at T_0 and T_1, and x quality of steam at the end of adiabatic expansion (on the supposition of the steam being saturated and dry at the beginning). Then

$$\int_{T_0}^{T_1} k\frac{dT}{T}+\frac{L_1}{T_1}=\frac{L_0}{T_0}x \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [1]$$

This, calculated by means of the tables of latent heat of Regnault or Henning, with average value $k=1.025$, gives

$$x=0.797 \quad \text{and} \quad \eta=\frac{Q_0}{Q_1}=0.266$$

for the first case.

For the second case let T_1 be the temperature at which the steam is produced (100 deg. cent.), and T_2 (200 deg. cent.) the temperature at which it is introduced into the cylinder; C specific heat of steam under one atmosphere pressure. Then

$$\int_{T_0}^{T_1} k\,\frac{dT}{T}+\frac{L_1}{T_1}+\int_{T_1}^{T_2} C\frac{dT}{T}=\frac{L_0}{T_0}x \ldots\ldots\ldots\ldots\ldots\ldots 2]$$

Or, with $k=1.01$, and $C=0.46$,
$x=0.996$ and $\eta=0.103$

or less than half the efficiency of the cycle without superheat. The efficiency of the corresponding Carnot cycle is 0.296; the Rankine cycle is below it by about 10 per cent. and the cycle with superheat by about 65 per cent. The inferiority of the last is explained by an almost total absence of condensation in the cylinder, which results in $0.2\times563=112$ calories escaping transformation into work. In a practical case, steam is produced saturated at 160 deg. cent. (320 deg. fahr.), and superheated to 300 deg. cent. (572 deg. fahr.). with the condenser at 40 deg. cent. (104 deg. fahr.). Equation [2], with $k=1.01$ and $C=0.50$. gives $x=0.875$, and $\eta=0.265$, instead of 0.24, as in the case of saturated steam, thus showing an improvement in efficiency of about 10 per cent. The improvement in the case of the Carnot cycle would be, however, from 0.277 to 0.454, or 64 per cent. The main part of the rest of the paper has been already reported in THE JOURNAL, September 1912, p. 1392, from an article by the author in another periodical.

WATER-TUBE BOILERS WITH AUTOMATIC UNDERFEED FURNACES (*Chaudières aquatubulaires avec foyers automatiques à propulsion inférieure. Le Génie Civil*, vol. 61, no. 21, p. 413, September 21, 1912. 6 pp., 15 figs. and one plate of drawings. *d*). Description of the Sulzer *water-tube boilers and automatic underfeed stokers*, with detail drawings.

ON THE INFLUENCE OF THE MATERIAL AND THICKNESS OF WALLS OF ECONOMIZER PIPES ON THE UTILIZATION OF HEAT (*Über den Einfluss des Materials und der Wandstärke von Ekonomiserrohren auf die Wärmeausnutzung*, Arthur Hanff, *Braunkohle*, vol. 11, no. 25, p. 389, September 20, 1912. 1½ pp. *p*). Discussion of the problem *whether or not wrought-iron pipes* are *preferable to cast-iron ones in economizers* as far as the utilization of heat is concerned; cast-iron pipes being otherwise preferable because they are easier to clean. The coefficient of heat transmission between the gases and water in the economizer is

$$k=\frac{1}{\frac{1}{a_1}+\frac{1}{a_2}+\frac{\delta}{\lambda}}$$

where a_1 is the coefficient of heat transmission between the flue gases and the external wall of the pipe; a_2 same between the inner wall of the pipe and the feedwater; δ thickness of the metal wall in meters; λ coefficient of heat transmission of the wall material. The material and thickness of the wall affect therefore only the single factor $\frac{\delta}{\lambda}$. To show of how little importance this factor is, the author gives the following calculation:

For both cases: $\alpha_1 = 16.142$, $\alpha_2 = 660$. For cast iron $\delta = 10.5$ mm (say 0.4 in.), for wrought iron, 2.5 mm (or 0.1 in); the most advantageous coefficient of heat transmission $\lambda = 70$ is accepted for wrought iron, and the least advantageous $\lambda = 40$ for cast iron. Substituting these values in the above formula, the author obtains for wrought iron $k = 15.755$, and for cast iron $k = 15.70$, showing that, even with the above assumptions, cast-iron pipes are practically just as good as wrought-iron pipes as far as transmission of heat is concerned.

EXAMINATION OF THE METHOD OF MR. J.-PAUL CLAYTON FOR THE EXPERIMENTAL STUDY OF RECIPROCATING STEAM ENGINES (*Examen de la méthode de M. J.-Paul Clayton pour l'étude expérimentale de la machine à vapeur à piston*, V. Dwelshauvers-Déry. *Revue de mécanique*, vol. 31, no. 2, p. 105, August 31, 1912. 6 pp., 2 figs. *t*). The author considers Mr. Clayton's method highly valuable, especially for practical purposes. He calls attention to the article which he and A. Duchesne published in the *Revue de mécanique*, July 1901, in which they attempted to show experimentally that the laws of expansion and compression differed from one another. This ought to be expected since during expansion the metallic walls of the cylinder furnish heat to the working fluid, while during compression, or at least a large part of it, it is the superheated fluid that gives up its heat to the surrounding metal.

All the operations and measurements for this article have been made with extreme precision, with the values of all the ordinates and abscissae of the indicator diagrams, and their logarithms given. It is therefore easy to construct a logarithmic diagram. For the expansion the logarithmic diagram *resembles* a straight line, but is not; for the compression the curve is so pronounced, especially towards the end, that there is no possibility of mistaking it. A beginning of a loop appears at the end of compression of the *PV* diagram. Some attributed it to leaks, the author to condensation, and G. Duchesne proved definitely that it was so by compressing in the same cylinder and under the same conditions, incondensible air, and obtaining no loop. *The law of compression of steam in our engines differs therefore from that of expansion.*

As to the practical application of Mr. Clayton's method, the author draws attention to the necessity of taking the greatest precaution for obtaining correct indicator diagrams when they are to be used in this connection. As to the volumes indicated by the *abscissae* of the diagrams: *stroke reducers* with long cord must be absolutely avoided owing to the possibility of whipping: it is essential that when the piston is at exactly 0.1, 0.2, etc. of its stroke, the pencil should be at exactly the same points of *its* stroke, which cannot be done if the stroke reducer consists of a cord which shortens and lengthens, or if the rigid joints have a play. The *clearance* must also be exactly determined either by calculation, or by filling it with water, and still better in both ways. One must not rely too implicitly on the data furnished by the manufacturers as to the ordinates: it is safer to have the instrument calibrated personally. Usually an *average diagram* is made of the head and crank diagram, which is used as if it were a real diagram. This of course is wrong; the average value of X_e is therefore wrong, although it is very difficult to say just how far it is wrong. The only way to obtain the most exact data possible is to use a surface condenser at each end of the cylinder. The author found in some of his experiments that even with separate valve gears for both sides of the cylinder, and everything, including clearances, on both sides exactly similar, there were still differences between the head and crank diagrams, due probably to different speeds of the piston at both ends of its stroke. The two condensers method is therefore the only way to obtain really exact data.

Manner of Working of Steam Generators (*Innere Arbeitsweise von Dampfgeneratoren*, B. Schapira. *Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 35, no. 37, p. 388, September 13, 1912. 2 pp., 9 figs. *gt*). General discussion of the *process of steam generation in a boiler*, mainly with regard to its dependence on the internal circulation.

Purification of Water by Permutit (*Épuration des eaux par la Permutite*, L. Clera, *Portefeuille économique des machines*, ser. 5, vol. 10, no. 681, p. 130, September 1912. *g*). A general article on the use of permutit, without data of any specific tests.

New Arrangement of the Sulzer Heating Boilers (*Neue Anordnung von Sulzer-Heizkesseln für ausgedehnte Heizanlagen. Dinglers polytechnisches Journal*, vol. 327, no. 37, p. 590, September 14, 1912. 2 pp., 3 figs. *de*). Description and data of tests of the Sulzer *boiler for house heating*. Tests made by the Swiss Association of Steam Boiler Owners have shown an efficiency as high as 86.2 per cent, and an average flue gas temperature, even with the boiler overloaded, below 200 deg. cent. (392 deg. fahr.).

Means for Removing Scale in Boilers (*Über Mittel zur Bekämpfung des Kesselsteines*, E. E. Basch, *Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 35, no. 38, p. 393, September 20, 1912. 2 pp. *p*). Summary of recipes for the prevention of scale in boilers patented at various times in Germany, with short criticisms. Covers practically the same ground as the article reported in The Journal, February 1912, p. 309.

Testing of Materials and Materials of Construction

Experiments on the Influence of Width in Notch Shock Tests (*Versuche über den Einfluss der Breite bei Kerbschlagproben*, R. Baumann, *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 33, p. 1311. August 17, 1912, 4 pp., 10 figs. *e*). The International Association for Testing Materials specified the *width of test bars* for notch shock tests as 30 mm, but for plates decided to consider the thickness of the *plate* as the width of the bar. The width in this case is therefore variable, and its influence on the test results had to be determined. This was partly done by Ehrensberg, and more fully by the author as reported in the present article. The following are the most interesting conclusions arrived at: (*a*) in boiler plates after annealing the work required for rupture has decreased 116 per cent in case of "broad" plates, and increased 11 per cent with "narrow" plates; (*b*) in the case of square iron 40 mm on a side the work required for rupture suddenly decreased when the width passed 1.5 cm. The material which appeared as tough when not broader than 1.5 cm, acted as brittle in broader test pieces. The author explains it by the supposition that a wide piece does not permit the contraction of area and flow of material, and that the change of state occurring in broad pieces is entirely different from that in narrow.

The author thinks therefore that the specifications of the International Association for Testing Materials ought to be revised, and a narrower bar specified. The article contains full data of the tests, as well as data of tests on cast iron.

USEFUL EXOTIC WOODS AND THEIR APPLICATION FOR TECHNICAL PURPOSES (*Exotische Nutzhölzer und ihre Verwendung in der Technik, Th. Wolff. Prometheus*, vol. 23, nos. 1191 to 1193, pp. 742, 759 and 774, August 24, 31 and September 7, 1912. 12 pp. *dp*). Discussion of the application for technical purposes and description of properties of exotic, i.e. *non-european woods*, viz. quebracho (Argentinian wood, extremely hard, unaffected by air or water, good for railway ties, telegraph poles and bridges); teak; pock (has the greatest specific weight, extremely hard); hickory; pitch-pine; redwood; eucalypt; Australian cypress and blackwood; mahogany; cedar and cypress. The article contains only general information without test data.

ACTION OF ELECTRICITY ON REINFORCED CONCRETE (*Les effects de l'électricité sur le béton armé*, F. H. *Le Génie Civil*, vol. 61, no. 20, p. 396, September 14, 1912. 4 pp., 23 figs. *e*). Description and data of tests on the *action of electricity (stray direct currents) on reinforced concrete* made at the Technical School at Darmstadt, Germany, and University of Vermont in this country. Only the former will be reported here; for the latter see *Engineering News*, July 1911. With concrete blocks placed in air on a layer of moist sand, or in sweet water, and subjected to the passage of a current of 0.1 ampere at 140 volts, fissures appeared after a period of from 50 to 110 days, the location of the fissures depending entirely on the position of the anode; in blocks placed in salt water they appeared at the end of from 20 to 70 days, and the concrete was colored in the neighborhood of the fissures first greenish gray (ferrous oxide), and after some exposure in the air, brown (ferric oxide). Fissures appeared in blocks immersed in chalk water in about the same time. A series of experiments led to the belief that these fissures were produced not by an evolution of gases, but by a thickening of the iron anode enclosed in the concrete, owing to superficial oxidation. By further tests it was found that the resistance of the reinforcing members to being torn out is increased by the action of the current, which may be explained by the formation of a layer of oxide adhering to the concrete more strongly than the original clean iron.

CONTRIBUTION TO THE PROBLEM OF MECHANICAL TESTING OF SOFT RUBBER (*Beiträge zur Frage der mechanischen Weichgummiprüfung*, K. Memmler. *Mitteilungen aus dem Kgl. Materialprüfungsamt zu Gross-Lichterfelde West*, vol. 30, no. 3, p. 139, 1912. 11 pp., 10 figs. *e*). Description and data of tests to determine the *resistance of soft rubber to compression stresses* *and wear*, such as it is subject to when used in automobile tires. For previous tests of tensile strength of rubber by the same institution see THE JOURNAL, March 1912, p. 430.

EXPERIMENTS ON REDUCTION OF STRESS BY ROUNDING OFF SHARP ANGLES (*Versuche über die Spannungsverminderung durch die Ausrundung scharfer Ecken*, E. Preuss, *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no 34, p. 1349, August 24, 1912. 5 pp., 28 figs. *e*). Description of and data from experiments made at the testing laboratory of the Darmstadt Technical High School on the *reduction of stresses* in metal members *by rounding off sharp angles*. The author states that although the fact that the stresses are particularly large at sharp angles was known for a long time, there are practically no experiments available showing numerically the precise influence of the radius of curvature on the stress at and near the point of flexure. Fig. 6 shows graphically how the change in the radius of curvature affects the distribution of lines of equal stresses. Among other things, these experiments have established the fact that the neutral point of stresses as found from actual measurements does not coincide with that determined theoretically. A complete account of these experiments will appear shortly in the *Mitteilungen über Forschungsarbeiten*.

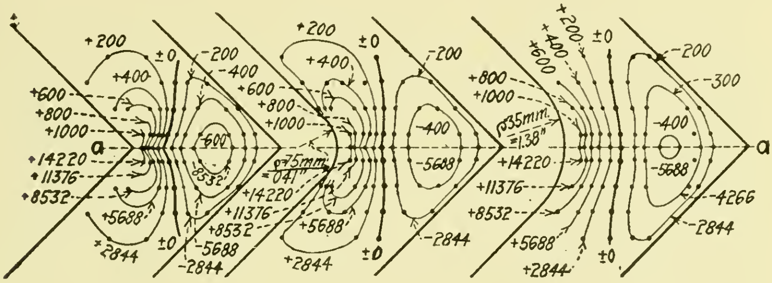

FIG. 6 DISTRIBUTION OF LINES OF EQUAL STRESSES FOR VARIOUS RADII OF CURVATURE
+Tensile Stress; − Compression; 0 Points constructed from Measurements: above Line *aa* in kg/qcm, below in Lb. per Sq. In.

USE OF TRASS IN REINFORCED CONCRETE (*Gebruik van tras bij gewapend beton*, E. Jacobs. *De Ingenieur*, vol. 27, no. 26, p. 734, September 7, 1912. 4 pp., 4 figs. *eA*). Data of tests for determining the influence of the proportion of trass in reinforced concrete on the properties of the latter. The tests were made with "Heidelberg" portland cement and Andernach trass to supplement the tests of Professor Van der Kloes (presented at the Copenhagen congress of the International Association for Testing Materials). The author concluded that in beams with maximum 2 per cent reinforcement (which is seldom exceeded) concrete with say ¾ portland cement, ¼ trass, 2 sand, and 2 gravel is in general just as strong as concrete with pure cement. The article gives full data of the tests, and drawings of stressed sections of the beams tested.

Determination of the Proportion of Original Ingredients in Cement, Mortar and Concrete after their Setting (*Die Bestimmung des Mischungsverhältnisses von abgebundenem (erhärtetem) Zementmörtel und -beton*, H. Burchartz. *Mitteilungen aus dem Kgl. Materialprüfungsamt zu Gross-Lichterfelde West*, vol. 30, no. 3, p. 117, 1912. 4½ pp. *ep*). It is sometimes necessary to determine the proportion of original ingredients in mortar or concrete after their setting, e.g. in case of an accident to a building when it is desired to find out whether or not it was due to poor mixture. The present article indicates the conditions when such a determination can be made, and the methods of procedure.

Tests of Slag Cement for the Purpose of comparing it with Portland Cement (*Prüfung von Eisenportlandzement im Vergleich zu Portlandzement*, M. Gary and H. Burchartz. *Mitteilungen aus dem Kgl. Materialprüfungsamt zu Gross-Lichterfelde West*, vol. 30, no. 3, p. 122, 1912. 8 pp. *ce.*) Data of supplementary tests for the determination of the properties of *slag cement* as compared with portland cement. For the main investigation see The Journal, May 1912, p. 805.

Action of Oil Concrete and Reinforced Concrete (*Einwirkung von Oel auf Beton oder Eisenbeton. Deutsche Bauzeitung*, no. 11, 1912, through *Metall-Technik*, vol. 38, no. 35, p. 288, August 31, 1912. ½ p. *p*). From a symposium of answers to a general inquiry sent out by the German Cement Association to its members. The action of *fat oils*, i.e. animal or vegetable, is different from that of mineral oils. The former become "rancid," i.e. decompose into glycerin and acid of fat, the latter entering into chemical composition with the calcium of the cement and thus decomposing it, which is especially noticeable in the case of porous mortars. In practice, machinery foundations subject to the action of constantly dropping lubricating oils and grease, are gradually affected and destroyed. Cement tanks for oil are sometimes made useless after a service of a couple of months. On the other hand *mineral oils* form no acids having affinity to calcium, and therefore do not affect cement at all.

Previous Treatment of Sheets before their Use in Boiler Shops (*Traitement préalable des tôles avant leur emploi en chadronnerie*, Desjuzeur. *La Houille blanche*, vol. 11, no. 7, p. 178, July 1912. 8 pp., 5 figs. *epA*). Investigation of the problem of *rendering boiler sheets* by a previous treatment *less sensible to irregularities of working* during the manufacture of boilers. The author does not reach any final conclusions, but shows that boiler sheets hardened, tempered and then cold worked, are in about the same condition as sheets first worked and then tempered; this may be of some value because it is probably easier to harden and temper the sheets first rather than temper them after cold working, although the cost of hardening must also be considered. The second part of the article contains data of a series of interesting tests the purpose of which was to determine whether *punching of rivet holes* in boiler plates produces only a permanent deformation in the neighborhood of the hole, or starts cracks as well, and whether this may be best remedied by annealing the boiler plate, or by reaming out the punch hole, and in the latter case, how much must be reamed out. Tests were made with plates of various thicknesses, and it was found (full data are given in the article) that plates 10 mm (say 0.4 in.) thick not annealed after punching cracked after bending through 53 to 59 deg., while the same plates punched and subsequently annealed could be bent through 110 deg. without cracking; previous hardening and annealing makes it comparatively safe to bend plates with punched and reamed out holes, but does not permit of bending plates with holes simply punched, and cannot therefore be substituted for reaming. Plates 15 mm (say 0.6 in.) thick, when simply punched, cracked after bending through 107 deg., and broke when bent to 131 and 147 deg., but plates with drilled, or punched and reamed, holes (even reamed only to 2 mm, or 0.078 in.) could be bent with only slight flaws at the edges. Annealing had the same effect as reaming, in permitting plates to be bent without cracks. In general it appears that reaming of only 2 mm (0.078 in.) in diameter removes the part of the metal altered by punching, and that annealing is also effective in restoring to the metal its original ductility, thus proving that punching produces a permanent deformation of the metal, and not cracks, since in the latter case annealing would do no good. But though more rapid and probably less expensive than reaming, annealing has the disadvantage of tending to change the shape of the plate, especially if it is thin, and to increase the non-coincidence of holes in the assembled boiler shell; besides, it does not remove the burr of the punch which is in the way of a good contact at riveting. Reaming seems to be better in every way, but if only of 2 mm (0.078 in.) in diameter, though sufficient according to the author's tests to remove the permanently set metal, may not be large enough in the case of superimposed plates, owing to a possible non-coincidence of the punched holes; since the divergence of the holes may be normally supposed to amount to about 1 mm (0.039 in.), the hole ought to be reamed to 4 mm (0.16 in.) so as to be sure to take care of the eccentricity of the hole in the lower plate.

Finally the author reports some interesting expanding tests on boiler sheets, which though not directly connected with the above tests, prove the importance of reaming holes after punching. The tests were made on a plate from a shell of a water conduit under pressure which was broken in an accident to the conduit. Rings were cut out from the plate, 100 mm (3.9 in.) in diameter, with the hole in the center and submitted to the expanding test. The best showing a ring with a punched hole made was when a ring with a hole 22.5 mm in diameter broke after the hole was expanded to 30 mm, while rings with drilled holes made from the same sheet broke only after a hole 22 mm in diameter was expanded to 55 and even 69 mm in diameter, thus showing that a plate with a drilled hole is several times stronger than one with a punched hole.

Testing of Materials in a Modern Automobile Factory (*Materialprüfung in einer modernen Automobilfabrik, E. Renisch. Auto-Technik*, vol. 1, nos. 15, 16, 17 appended to *Allgemeine Automobil-Zeitung*, vol. 3, nos. 33, 35, 37. 8 pp., 46 figs. *dp*). Description of the testing laboratory of the Protos Automobile Company, Nonnendamm, Germany. In addition to the usual apparatus for mechanical testing, this laboratory is fitted with a complete installation for metallographic investigation and microphotography. In testing, particular attention is paid to the appearance of the surface of fracture, as well as to tests for homogeneity of structure of the metal in general, and in particular to the presence of segregated particles of sulphur and phosphorus. For the latter purpose the Heyn test is used: the piece of metal is ground and polished along and across the grain, and etched by suitable chemicals. For etching of a polished surface, see A. Martens and E. Heyn, Handbuch der Materialienkunde (Berlin 1912), vol. 2, pt. A, p. 178 ff.

Pure Zinc Pipes (*Rein-Zinkrohre, M. Haustechnische Rundschau*, vol. 17, no. 5, p. 51, September 1, 1912. 1 p., 15 figs. *d*). Description of *zinc* pipes of German manufacture for use in *house water piping.* The advantages claimed for them in comparison with lead and galvanized iron pipes are: they are less affected by water and its impurities than either of the latter; they do not give rise to local electric currents due to the contact of two different metals (in the case of galvanized iron pipes) and rapidly destroying the pipes; even if some zinc be dissolved by the water, it is harmless. Tests at the Royal Laboratory at Gross Lichterfelde West have shown that mechanically zinc pipe is much stronger than lead pipe. It also gives a very strong welded joint.

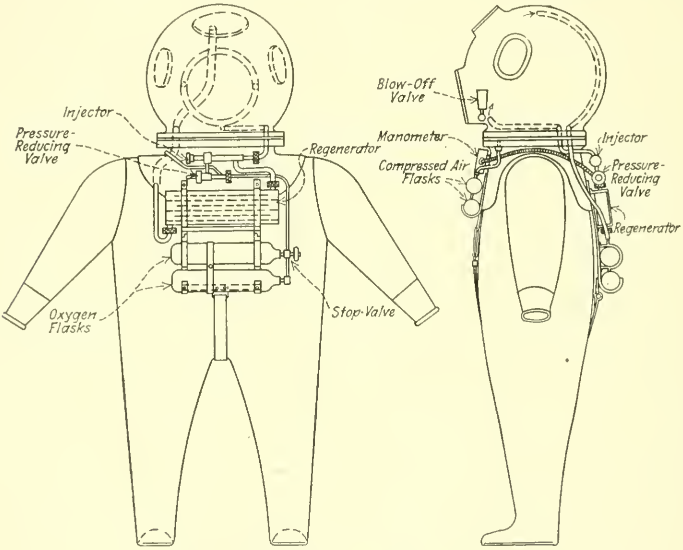

Fig. 7 Diving Apparatus Westfalia

Miscellanea

New Diving Apparatus Westfalia (*Die neuen Tauchgeräthe "Westfalia."* Grahn. *Glückauf*, vol. 48, no. 36, p. 1448, September 7, 1912. 3 pp., 5 figs. *d*). By invitation of the German Navy Office several firms have endeavored to construct a *diving apparatus without air pipe*, such as might be used by the crew of a submarine in case of accident. Such an apparatus built by the Lubeck Dredging Machinery Company was described in THE JOURNAL, September 1912, p. 1584. The present article treats of one by the Westfalia Armaturen- und Maschinenfabrik in Gelsenkirchen, Germany. Both factories had, previous to the construction of diving apparatus, a large experience in the manufacture of similar apparatus for miners and firemen. As shown in Fig. 7 the diver carries on his back a regenerator tank taking care of carbon dioxide and water vapor from breathing, while below it are steel flasks with gas for breathing. This gas is not pure oxygen, which is poisonous when at high pressure (cp. *Deutsche medizinische Wochenschrift*, no. 32, 1912), but

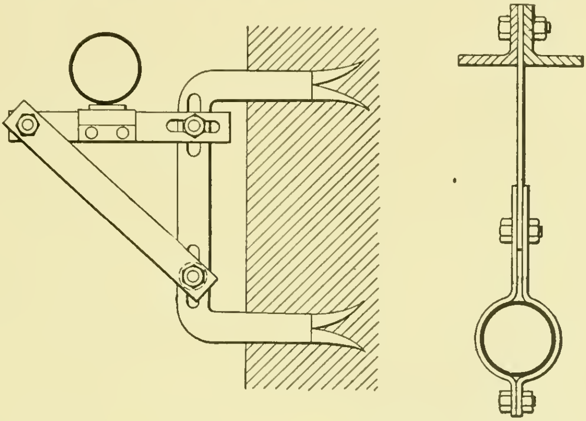

FIG. 8 FRANZ MAASS AND HARDT ADJUSTABLE PIPE SUPPORT

a mixture of oxygen and compressed air in such proportion that even at a depth of 50 m (164 ft.) corresponding to a pressure of 6 atmospheres absolute the contents of oxygen does not exceed 23 per cent. Various ratios of the mixture are used for work at various depths; thus, for 20 to 30 m (65 to 98 ft.) 45 per cent oxygen and 55 per cent nitrogen, while for depths from 50 to 60 m (164 to 196 ft.) 30 per cent oxygen and 70 per cent nitrogen. The oxygen and air or nitrogen are not kept in separate vessels, but the flask is first filled with compressed air up to a certain pressure, and the oxygen is then pumped in, special tables being used for the determination of the pressure corresponding to certain percentages of oxygen in the mixture. The pressure-reducing valve between the steel flasks and injector is so arranged as to vary automatically the amount of gas admitted in proportion to the water pressure, the diving outfit being thus always filled and permitting of easy and comfortable breathing. The compressed gas mixture serves also to drive the injector which, with the help of suitably arranged piping, takes the vitiated air from the upper part of the helm and returns it thither after it has passed through the regenerator and become free of the carbon dioxide and moisture. On the front side are placed two flasks with compressed air, to be used for rapidly filling the diving suit when the diver wants to come to the surface quickly. The manometer permits him to see what supply of air and oxygen he still has at his command. The article also describes a second diving outfit without helm or diving suit, for use for short stay under water or in case of emergency, e.g. accident to a submarine.

LIQUEFACTION OF HYDROGEN WITH NO COLD LOSSES (*Verflüssigung des Wasserstoffs bei Vermeidung von Kälteverlusten*, K. Olszewski, *Krak. Anz.*, 1912. pp. 1 to 27, through *Beiblätter zu den Annalen der Physik*, vol. 36, no. 18, p. 969. *d*). Description of the Improved *Olszewski hydrogen liquefier, thermostat* for temperatures from -190 to 0.5 deg. cent., and gas liquefaction plant in Krakau, Austria.

POROUS METALS (*Les métaux poreaux*, H. I. Hannover, *Revue de métallurgie*, vol. 9, no. 9, p. 641, September 1912. 5 pp., 7 figs. *dt*). General remarks on the solidification of metallic alloys, and description of the author's method of producing porous metals, chiefly lead plates for accumulators (eutectic alloys).

ADJUSTABLE PIPE SUPPORTS (*Verstellbare Rohrträger*. *Prometheus*, vol. 23, no. 1194, p. 799, September 14, 1912. 1 p., 5 figs. *d*). Description of an *adjustable pipe support* to be used either when the position of a pipe has to be changed, or when the original support has been placed incorrectly. By unscrewing the two bolts, the pipe may be adjusted for practically any desired position within a fairly wide range of variation (Fig.8).

DREDGES (*Les dragues*, A. Baril, *Revue de mécanique*, vol. 31, no. 2, p. 111, August 31, 1912. 43 pp., 82 figs. *d*). Description of dredges, mainly for harbor work, European types.

Supplementary References

PREHEATING THE AIR IN INTERNAL COMBUSTION ENGINES (April and May, 1912, pp. 615 and 793). Cp. another report of same article in *Power*, October 15, 1912, p. 565, and an article on the same subject in *Gas Engine* (Preheating Gas Engine Charge, no. 10, 1912, p. 538).

THEORY OF STEEL-WELDING (June 1912, p. 961). The same article reported in *Acetylene Lighting and Welding Journal*, September 1912, p. 205.

SOMETHING ABOUT DIVING (October 1912, p. 1584). For a description of the Lubeck diving apparatus see also *Scientific American*, October 5, 1912, p. 284.

REPORTS OF MEETINGS

MEETINGS ON SMOKE ABATEMENT

The problem of Smoke Abatement has been considered at two recent meetings of the Society, George H. Perkins, head of the engineering department of the Lowell Textile School, giving his report on the International Smoke Abatement Exhibition and Conference, held in London in March and April 1912, at which he represented the Lowell Textile School and also the Society.

At the first of these, a joint meeting of the Philadelphia Section and the Engineers Club of Philadelphia, held in that city on October 5, a paper on Methods and Means of Smoke Abatement, was presented by the author, O. R. McBride of Pittsburgh, following Mr. Perkins' report, and was profusely illustrated with lantern slides. Talks on the sanitary aspect of the smoke problem were given by Dr. Andrews Allen, representing the County Medical Association, and Dr. Anders, both tracing the relation between the presence of smoke in the atmosphere and the spread and virulence of disease. John M. Lukens, chief of the Bureau of Boiler Inspection of Philadelphia, gave a brief sketch of the municipal efforts in the direction of smoke abatement in that city, and quoted figures to show that about 500 smoke-consuming appliances have been installed there. Henry S. Morris recommended for this purpose the wider use of gaseous fuel as both a cleaner and on the whole a more economical method than burning coal in the furnace. Henry J. Hartley, on the other hand, ascribed a large part of the smoke nuisance to the desire of manufacturers to get out of a boiler more than its capacity, and to their trying to force through more fuel than the grates can take care of properly. W. C. Furber said that smoke inspectors are on duty in Washington until six o'clock each evening and that the hotels may smoke as much as they want to after that hour. Prof. Robert H. Fernald insisted on the necessity of eliminating the political factor from the problem of smoke prevention, and of exceedingly careful study of the type of furnace and other conditions. In conclusion he made favorable mention of the work done in this direction in Cleveland.

At a meeting of the members in New York and vicinity, held on October 8, more than 300 members and representatives of various municipal and technical organizations were present. Following Mr. Perkins' report, Paul P. Bird told how the smoke problem is being solved in Chicago and what the smoke department there is doing in this respect. Prof. O. P. Hood, representing the United States Bureau of Mines, followed him with a statement of the attitude of the Government on this problem. Lieut. James Reed, formerly assistant director of Public Works of Philadelphia, and John M. Lukens, pointed out the important role of the railroads in this respect.

The scientific side of the problem of smoke prevention was taken up by Prof.

L. P. Breckenridge of Yale University, who concluded by stating that with the data now available, and knowing the volatile content in a coal and rate of combustion, it is entirely possible to design furnaces properly. The same line of discussion was continued by Professor Kent and Dr. Jacobus, who showed that smoke may be prevented and explained the difficulties in the way of its complete elimination. Mr. Adler closed the discussion by a review of the principles of combustion, demonstrating that the essential conditions of smoke abatement are a long flame length, more refractory and better firebrick, and a high chimney.

Written discussions were submitted by Charles T. Main, Horace C. Porter of the Bureau of Mines, Pittsburgh, Prof. J. A. Moyer, Perry Barker, H. V. O. Coes, Dean C. H. Benjamin, E. C. Fisher, C. S. Davis of the William B. Pierce Co., Buffalo, N. Y., George R. Henderson, J. S. Shoemaker of Cumberland Mills, Me., E. A. Uehlung, and M. C. Huyette of Murphy Iron Works, Buffalo, N. Y.

BOSTON MEETING, OCTOBER 16

The first meeting of the season in Boston was held on October 16, the Boston Society of Civil Engineers and the Boston section of the American Institute of Electrical Engineers coöperating with the Society. A paper on the Water Power Development of the Mississippi River Power Company at Keokuk, Iowa, was presented by Hugh L. Cooper, chief engineer of the company. By way of introduction Mr. Cooper showed a few slides illustrating notable features of other undertakings. The immense undertaking at Keokuk and its several parts in various stages of progress were very fully illustrated and described.

In the discussion of the paper which followed, E. L. Galusha, engineer with the Stone and Webster Engineering Corporation of Boston, presented an illustrated description of the main electrical features of the plant, including the generators, switching and other power-house equipment, and transmission line construction.

NECROLOGY

ALBERT W. DANFORTH

Albert W. Danforth who died at his home in Lowell, Mass., on April 25, 1912, was educated in the Lowell public schools and began his professional career as mechanical engineer for the Lawrence Corporation of that city. A few years later he became mill and constructing engineer for the Hamilton Corporation, and subsequently designer and superintendent of construction on machinery for the United States Cartridge Company, and mill and mechanical engineer in Newburyport. In 1882 he went to China in the employ of the Chinese Government to build a cotton mill at Shanghai, owned by Li Hung Chang, then viceroy of the province, remaining there for twenty years and doing much to introduce European methods of cotton manufacture into the country.

SAMUEL TENNEY MUDGE

Samuel Tenney Mudge died at his home in South Manchester, Conn., on September 8, 1912. Mr. Mudge was born in Brooklyn, N. Y., on February 22, 1883, and received his education at the Stevens Institute of Technology, from which he was graduated with the degree of M.E. in 1906. He entered the service of the American Sugar Refining Company in Jersey City in the same year, and two years later became instructor in machine design in the University of Michigan. At the time of his death he was general manager and superintendent of the mills of the Rogers Paper Manufacturing Company, South Manchester, Conn.

J. M. ROBINSON

J. M. Robinson, president of the Crex Carpet Company of New York, who died on March 30, 1912, was born in New York on July 19, 1849, and educated in the public schools. He received his engineering training in Cooper Institute and through practical experience.

Mr. Robinson had charge for some time of the linesmen and building of the Western Union Telegraph Company, and was associated with the Mills Estate, erecting the Mills Buildings in New York and San Francisco, and having active management of the property. He was also interested in many commercial enterprises, including the company of which he was president at the time of his death.

Mr. Robinson was a resident of Teaneck, N. J., and actively interested in the Board of Education and in political and social affairs of that township.

AMORY E. ROWLAND

Amory E. Rowland was born July 2, 1852, in Brooklyn, N. Y., and educated in the Sheffield Scientific School of Yale University. He gained experience after graduation in practical mechanics in the Continental Iron Works in Brooklyn and had been connected with the firm of F. C. & A. E. Rowland in New Haven and its successor, the Rowland Machine Company, about thirty-five years. At the time of his death on May 7, 1912, he was secretary and treasurer of the company.

Mr. Rowland was a trustee of the New Haven Savings Bank, director and prominently identified with the New Haven Colony Historical Society, the Graduates Club of New Haven, the Organized Charities Association, and other similar organizations.

JAMES B. SINER

James B. Siner, a mill engineer of Boston, died on September 17, 1912, at his home in Malden. Mr. Siner who was born in Lowell, Mass., on April 3, 1835, was educated in the public schools of that city and served his apprenticeship in the Lawrence Machine Shop. He began his career as mechanical superintendent of the Washington Mills at Lawrence, with which he was associated for twenty-two years, and was at the time of his death in charge of the appraisal department of the Factory Mutual Fire Insurance Company of Boston.

EMPLOYMENT BULLETIN

The Society considers it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is pleased to receive requests both for positions and for men. Notices are not repeated except upon special request. Names and records, however, are kept on current office list three months, and if desired must be renewed at the end of such period. Copy for the Bulletin must be in hand before the 12th of the month. The list of "men available" is made up from members of the Society, with names of other good men not members, and are on file in the Society office. Information will be sent upon application

POSITIONS AVAILABLE

0210 Assistant engineer, permanent position; must be experienced designer of electric traveling cranes, accurate and quick at figures, capable of systematizing engineering data. References as to experience and personal habits. State technical training and salary desired.

0211 Competent instructor in experimental mechanical engineering in one of the prominent state universities. Candidates for this position should have had successful teaching experience and some practical work. Salary available $1500 a year, with excellent prospects for advancement.

0212 Boiler shop superintendent or foreman to take charge of contract shop with blast furnace, smoke stack, tank and miscellaneous work. Live progressive man, but preferably with technical training, though not necessarily.

0213 Jig designer for New Jersey concern.

0214 Man to take charge of mechanical work of new plant, general millwright repairs, installation of additional machinery, repairs to motors and electrical machinery and the installation of new machinery; would have charge of small force of electricians and machinists, lay out work and keep up repairs in good shape.

0215 Leading draftsman on electrical work for prominent concern; conversant with power and substation equipment and capable of working up general plans and details of such work. Man of intelligence and initiative.

0216 Electrical engineer to act as associate to engineer in charge of electrical division of prominent Massachusetts concern; well grounded in theory and construction of such electrical apparatus as enters into station equipment and able to work out general problems connected with the extension of a plant, to make estimates, specifications, etc.

0217 Designer for rolling mill installations and similar types of heavy machinery. Must be a technical graduate. Salary $175 to $225.

0218 Opening in engineering department for a technical graduate of concern manufacturing ball bearings. Would want a man who would begin as draftsman and work up to an engineer's position. Location Connecticut.

0219 Concern in New York State desires services of man for the purpose of investigating the electrical power propositions of the company, especially as applied to low-pressure steam turbines driven from mill engine exhaust. Applicant would have to be experienced along this line of work and able to take up the entire question.

0220 Shop superintendent for large plant, manufacturing line of small specialties, located in New Jersey, about 20 miles from New York. Position will require a technical graduate having considerable experience particularly in the manufacture of articles requiring interchangeable parts, automatic machinery, etc.

0221 Designer to work out new design of jig or develop present design.

0222 Man with stationary or marine engineering experience, familiar with the combustion of oil and the analysis and control of the gases of combustion by means of the Orsat apparatus, pyrometers, etc. Location small New Jersey town, about 30 to 35 miles from New York City.

MEN AVAILABLE

518 Junior, mechanical engineer, Cornell, age 27, married, desires position of some responsibility in inspection or production department of progressive manufacturing concern; experience in foundry of prominent pump manufacturer; cost department of a recognized maker of machine tools; tool room of large experimental machine shop; now employed as foreman of automobile assembly. Satisfactory references as to character and ability.

519 Member, M.I.T. graduate, married, desires position as plant engineer, purchasing agent or superintendent of construction; 20 years' experience in estimates, valuations, purchases, designs, specifications construction, equipment testing and general operation of steam and hydraulic electric power plants, factories, textile mills and electric transmission of power. Salary $3600 to $4000.

520 Associate, age 35, desires position of responsibility in or near Philadelphia. Sixteen years' experience in civil, structural and mechanical lines, as draftsman, squad foreman, chief draftsman and checker at rolling, steel and pipe mills, blast furnaces, steel cars and locomotives, byproduct coke ovens and on chemical apparatus.

521 Technical graduate with degrees of M.E. and C.E., instructor for a number of years in machine design and descriptive geometry, wishes to take charge of machine design or descriptive geometry departments of a university or technical school. Author of recent text books on above subjects. Would accept position as mechanical or research engineer in manufacturing plant, consulting and testing work. At present employed but desires to change.

522 Recent graduate desires position as assistant to a superintendent or production engineer.

523 Member desires position as works manager, preferably with new

company just starting business; long experience as factory man and engineer; at present head of engineering department of large company.

524 Member, technical graduate, scientific training with thorough knowledge of chemistry and chemical processes, 9 years' experience in design and operation of steam and gas power installations and apparatus, specializing in the economic combustion and gasification of fuels, desires position with manufacturer; salary $3000; or will consider a position with consulting engineer on a salary partnership basis.

525 Mechanical engineer, thirty, desires responsible position with growing concern; nine years' experience in engine and boiler manufacture and general plate steel construction, in sales and engineering departments and as assistant manager; ability and initiative, good organizer. Graduate of leading engineering college. Eastern location desired, preferably Philadelphia, New York or Baltimore.

526 Member, at present at head of department of mechanical engineering of Eastern college, with thirteen years' successful experience as engineer of tests, mechanical engineer and superintendent of engine works, would like to become identified with firm of consulting engineers, or sales engineer with a manufacturing organization.

527 Sales manager, man of executive business capacity, 35 years of age, desires position. Technical education, manufacturing experience in foundry and machine shops, business experience, office and road.

528 Cornell graduate experience in hydroelectric, power plant and transmission line work.

529 Mechanical engineer, member, university graduate, 35 years' experience in the building, erection and managing of sugar and chemical plants, designing and building special apparatus, vacuum and condensing plants, power plants, etc., will consider position as superintendent, managing engineer of industrial plants, or take charge of branch office for the equipment of such plants.

530 Member, age 35, M.I.T. graduate, wishes to change. Fifteen years' experience in mechanical and civil engineering, including designing, erection, testing and operation and teaching engineering, five years in executive positions. Position as assistant to works manager of large plant or group of plants, or with consulting engineer. Minimum salary, $3000.

531 Manager or superintendent; mechanical engineer with practical shop, technical and commercial experience, expert in up-to-date manufacture of interchangeable parts, resourceful in design and process for reducing costs and increasing production, good organizer and production engineer with excellent executive ability. A-1 references.

532 M.I.T. graduate in mechanical engineering, age 25, has been employed as assistant to chief engineer in automobile plant. Desires position with concern having opportunity for such man, and where merit is determined by results.

533 Cornell graduate with several years' practical, mechanical, electrical and industrial engineering experience wants permanent position

with established company, with whom energy, perseverance and knowledge of up-to-date modern methods can be made of value. Thoroughly versed in design, development and building of special machinery and the management of men. Salary to be adjusted with value of services rendered.

534 Executive engineer $6,000 man open for position. Broad experience in design construction and operation of hydraulic, gas and steam power and manufacturing plants. Competent organizer and handler of men and machinery.

535 Junior member, three years' experience in selling, manufacturing and drafting. At present employed, desires position with manufacturing concern.

536 Graduate Massachusetts Institute of Technology, age 37, married, ten years' experience in pipe bending and coiling, also experience in feedwater heater and condenser design and in economical use of steam in manufacturing processes. Designing and selling automobile radiators and sheet metal parts.

537 Member, technical and practical mechanic, good executive with 12 years' experience as superintendent and manager; thoroughly posted on modern machine shop and foundry practice, desires position with live company. Experienced in steam engines, boilers, gas and gasolene engines, transmission and mill machinery.

ACCESSIONS TO THE LIBRARY

WITH COMMENTS BY THE LIBRARIAN

This list includes only accessions to the library of this Society. Lists of accessions to the libraries of the A. I. E. E. and A. I. M. E. can be secured on request from Calvin W. Rice, Secretary, Am. Soc. M. E.

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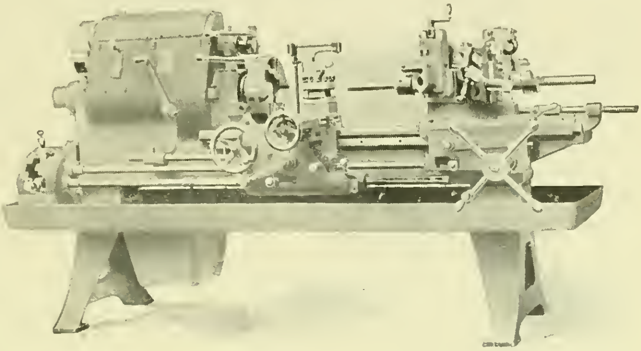

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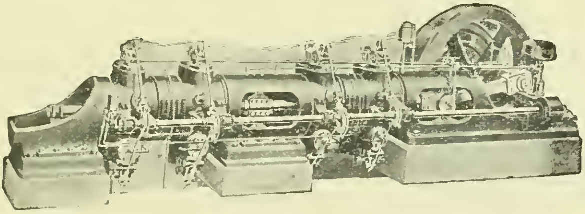

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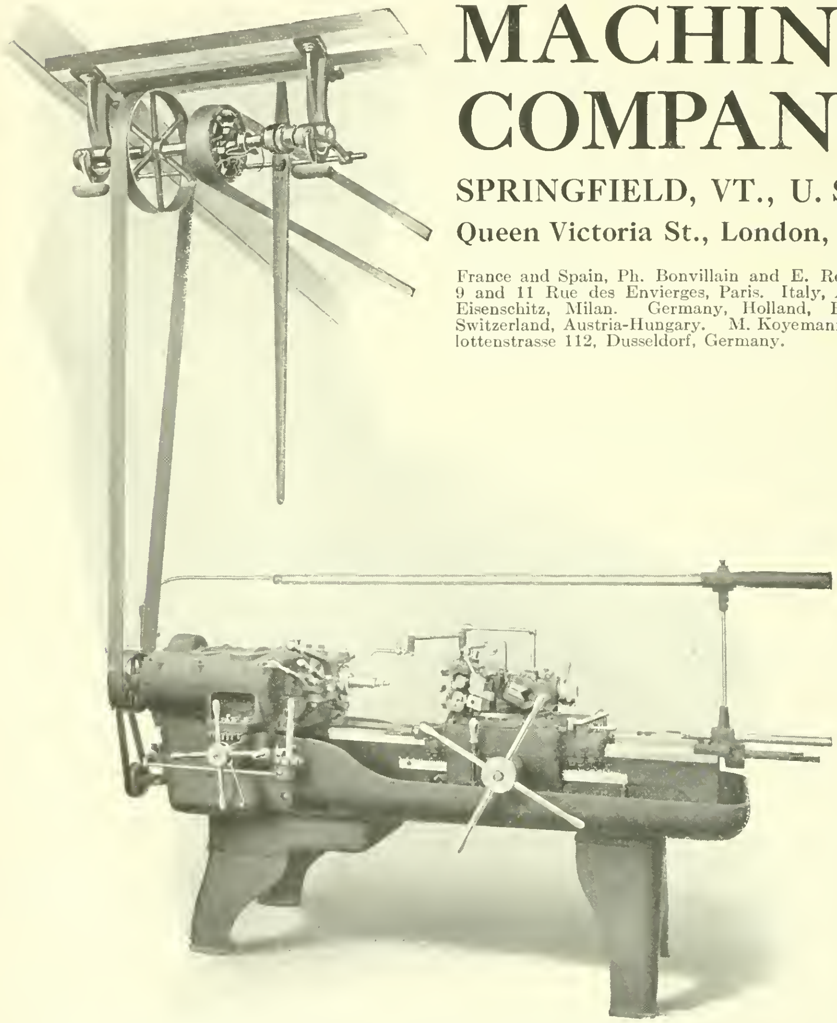

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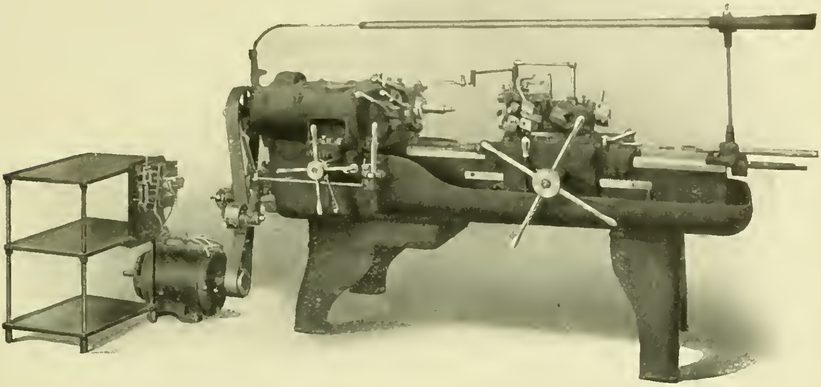

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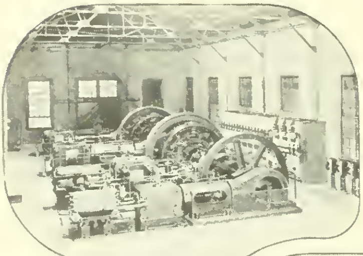

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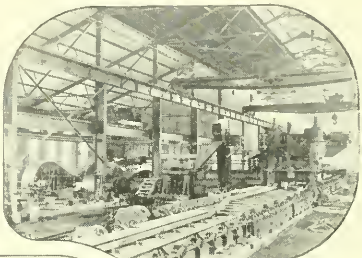

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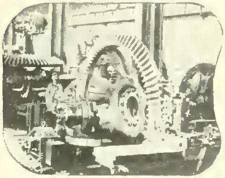

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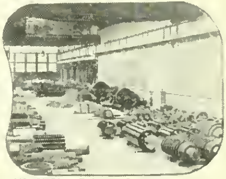

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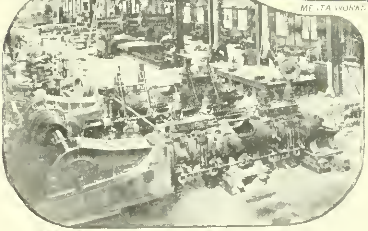

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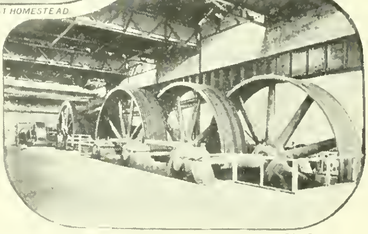

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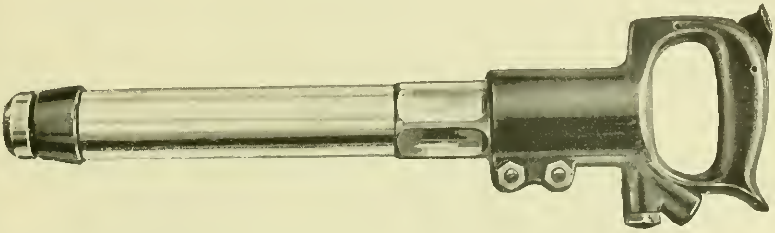

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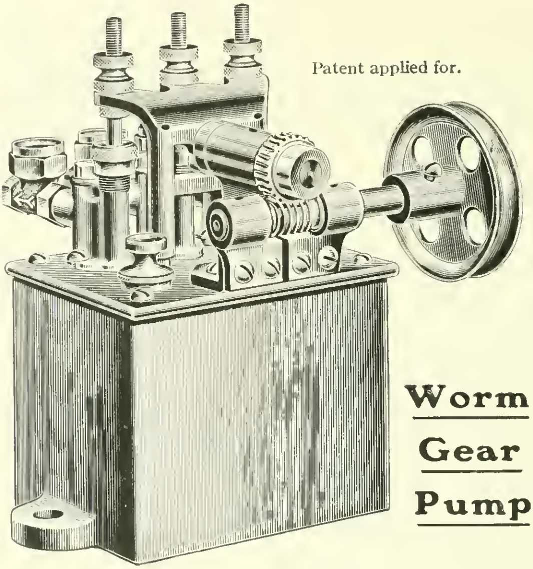

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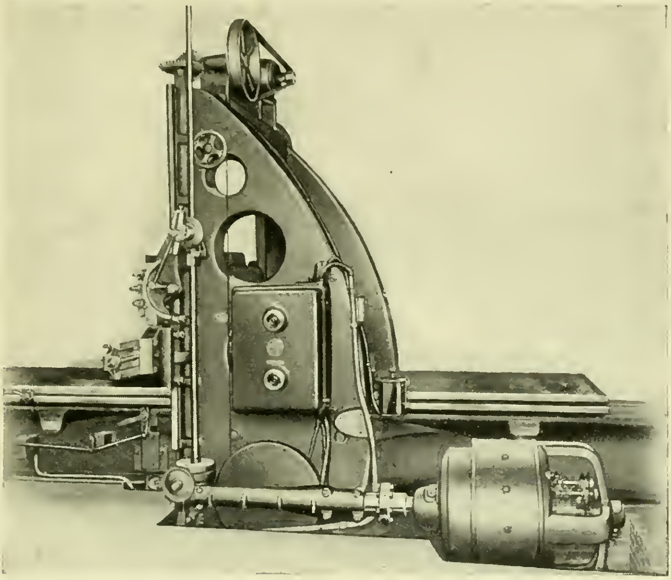

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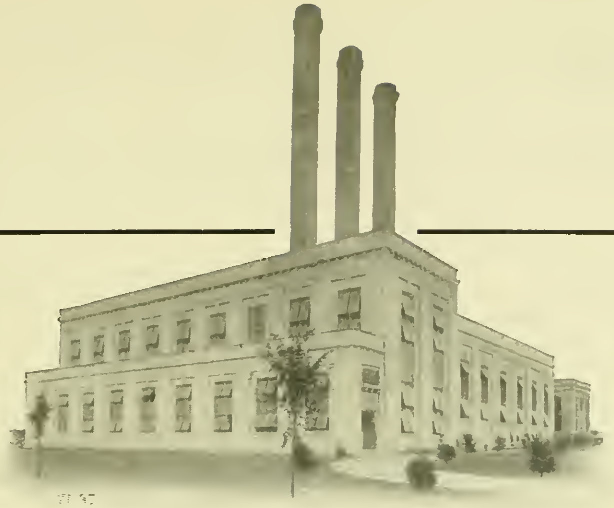

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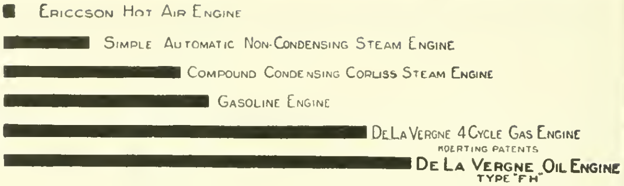

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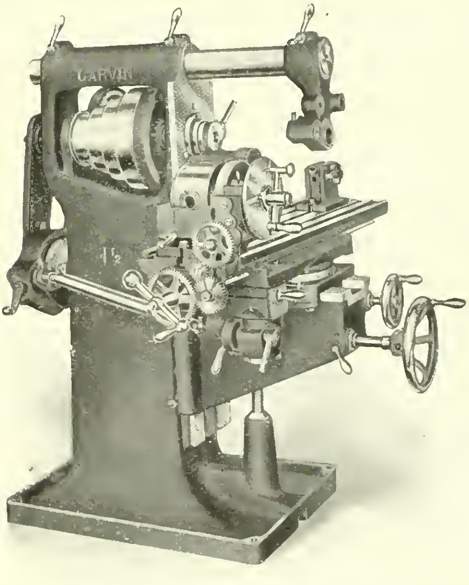

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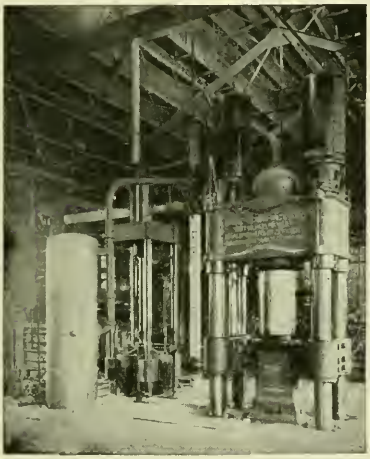

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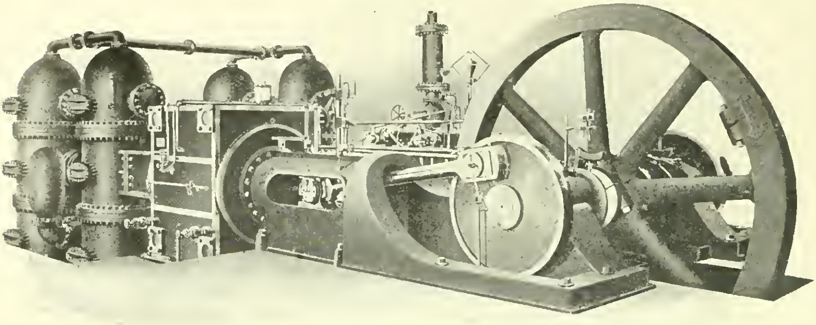

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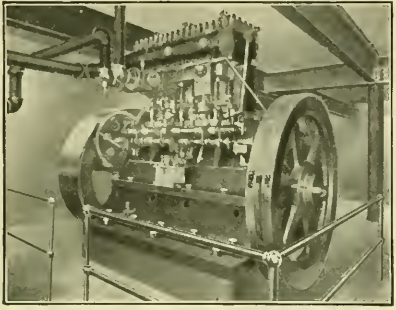

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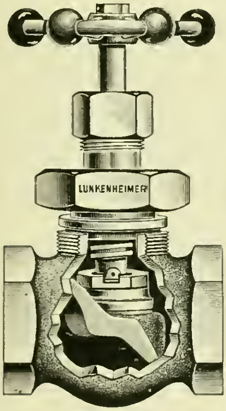

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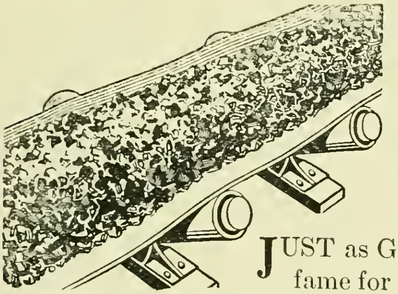

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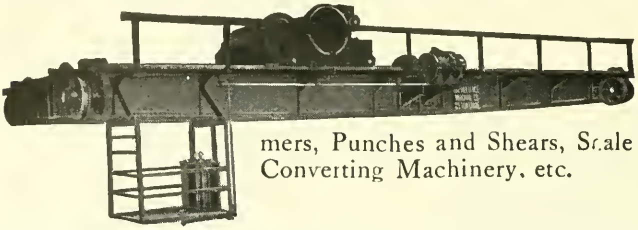

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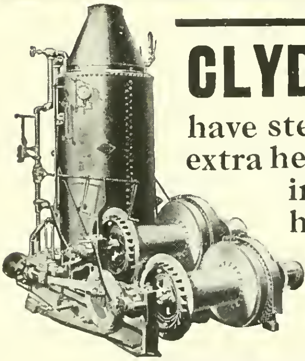

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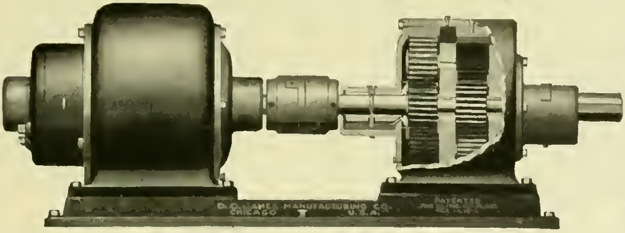

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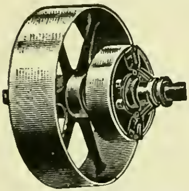
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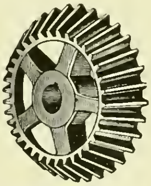
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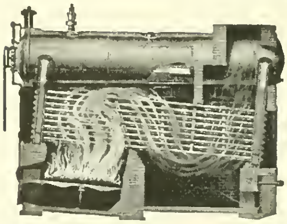

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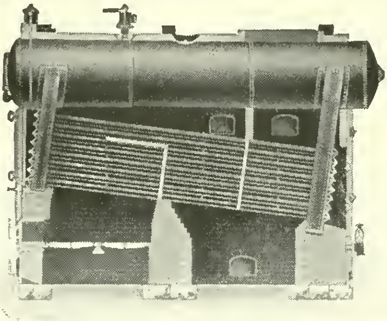

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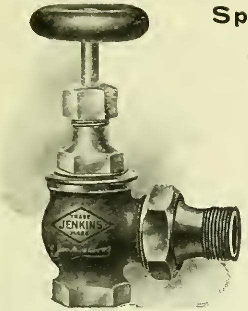

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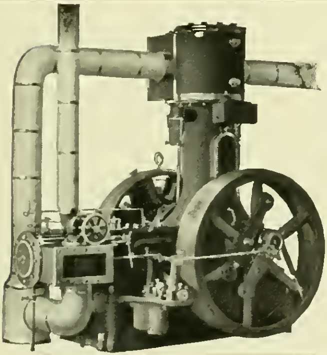

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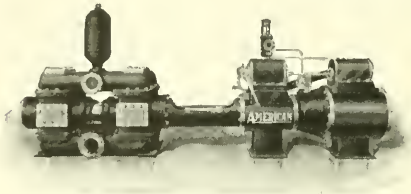

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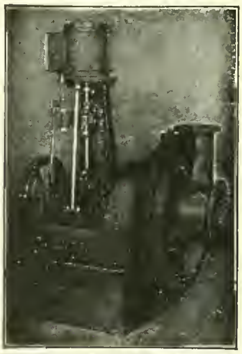

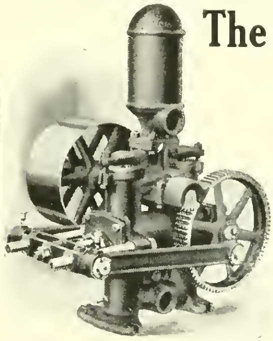

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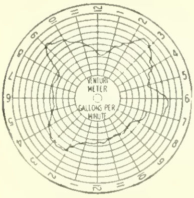

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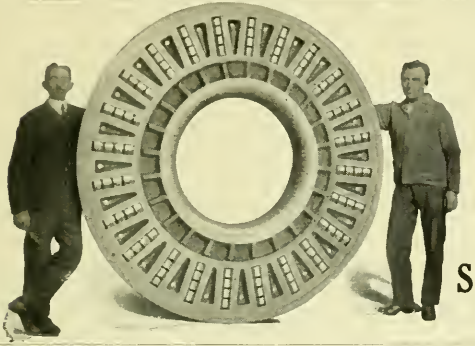

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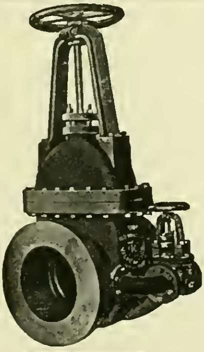

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THE JOURNAL

of

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

DECEMBER 1912

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MONTHLY MEETING: BOSTON, DECEMBER 20
MEETING IN GERMANY: JUNE 21–JULY 7, 1913

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Every member should not only make an especial effort to attend the convention, but should extend a cordial invitation on behalf of the Society to each of his engineering acqaintances. Guests of members will be welcomed even though the member extending the invitation finds it impossible to be present.

No member of the Society can fail to profit by the opportunity afforded for making new acquaintances as well as renewing old friendships, and by the many advantages which accrue from the mutual interchange of views on professional topics.

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DECEMBER 1912 VOL. 34 No. 12

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COMING MEETINGS OF THE SOCIETY

December 3–6, *New York City*, Engineering Societies Building, Annual Meeting. See page 3 for program.

December 20, *Boston, Mass.* Paper: Laboratory Experiences, Prof. H. S. Heywood of Massachusetts Institute of Technology.

June 21–*July* 7, 1913, Joint Meeting with Verein deutscher Ingenieure, Leipzig and various industrial cities of Germany. Party limited to 200 members and 100 ladies. See page 9 for tentative program.

THE JOURNAL

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOL. 34 DECEMBER 1912 NUMBER 12

ANNUAL MEETING

The Annual Meeting will be in progress as this number of The Journal reaches the membership. There are to be sessions under the direction of several of the sub-committees of the Committee on Meetings, by means of which a total of nine sessions have been arranged, and a greater variety of topics will be treated than at any previous meeting. In the last issue of The Journal a tentative program was published, with a partial list of the papers. The complete program is now given with the full list of papers and a statement of the various social events. In view of the fact that the date for the awarding of the John Fritz Medal by the four engineering societies is Thursday night of the convention, the Council have arranged for this event to take the place of the usual reunion.

There will also be the usual excursions, and a Ladies Committee has been formed to care for the entertainment of the visiting ladies.

PROGRAM

TUESDAY, DECEMBER 3

Opening Session, 8.30 p.m., Auditorium

President's Address, The Present Opportunities and Consequent Responsibilities of the Engineer. Alex. C. Humphreys

Report of Tellers of Election of Officers
Introduction of the President-elect

Reception

Reception by the President and President-elect and ladies, to the members and guests of the Society. A collation will be served at 10 o'clock.

WEDNESDAY, DECEMBER 4

Business Meeting, 10.00 a.m., Auditorium

Annual business meeting. Reports of the Council, tellers of election of membership, standing and special committees. Amendment to Constitution under C57, relating to election of members and membership grades. New business.

SIMULTANEOUS SESSIONS FOLLOWING BUSINESS MEETING

Power Plant Session, Auditorium

Dimensions of Boiler Chimneys for Crude Oil, C. R. Weymouth
Tests of a 1000-H.P. 24 Tubes High B. & W. Boiler, B. N. Bump
Air in Surface Condensation, George A. Orrok
Properties of Saturated and Superheated Ammonia, Wm. Earl Mosher
Physical Properties of Anhydrous Ammonia, L. S. Marks and F. W. Loomis

Presented through the courtesy of The American Society of Refrigerating Engineers

Experiments with North Dakota Lignite in a Steam Power Plant and a Gas Producer, Calvin W. Crouch
The Baltimore Sewerage Pump Valve, A. F. Nagle

This session if necessary will be continued in the afternoon

Textile Session, Sixth Floor

(Papers contributed by Sub-Committee on Textiles)

The Principles of Valuing Property, Henry K. Rowell
The Power Plants of Textile Mills, John A. Stevens

Gas Power Section, Sixth Floor

Transaction of business and election of officers
Chairman's Address, Recent Development of Gas Power in Europe, H. J. Freyn
Discussion by Wm. T. Magruder, D. G. Baker, and others
Heavy Oil for Use in Internal-Combustion Engines, Irving C. Allen

WEDNESDAY AFTERNOON

Reports of Technical Committees, 2.00 p.m., Auditorium

At this session will be presented reports of special committees, including Power Tests, Hoisting and Conveying, Standard Cross-Section Symbols, Involute Gears, Standardization of Catalogues, Flanges, Pipe Threads, and others.

WEDNESDAY EVENING

Reception and Dinner to Professor Sweet, 6.30 p.m., Fifth Floor

A reception and subscription dinner will be tendered to Prof. John E. Sweet, Honorary Member and Past-President, in celebration of his eightieth birthday and in recognition of his services to the engineering profession. Dinner at 7 o'clock.

THURSDAY, DECEMBER 5

Joint Session in charge of Sub-Committees on Machine Shop Practice and Iron and Steel, 10.00 a.m., Auditorium

(Papers contributed by Sub-Committee on Machine Shop Practice)

Report of Sub-Committee: The Development of Machine Shop Practice during the Last Decade

The Vauclain Drill, A. C. Vauclain and Henry V. Wille

Efficient Production of Cylindrical Work, C. H. Norton

Increase of Bore of High-Speed Wheels by Centrifugal Stresses, Sanford A. Moss

To be presented by title. Contributed through the courtesy of the Boston Local Committee

Investigation of Efficiency of Worm Gearing for Automobile Transmission, Wm. H. Kenerson

The Strength of Gear Teeth, Guido H. Marx

(Paper contributed by Sub-Committee on Iron and Steel)

Case Carbonizing, Marcus T. Lothrop

This session if necessary will be continued in the afternoon

Railroad Session, 10.00 a.m., Sixth Floor

(Papers contributed by Sub-Committee on Railroads)

Lighting of Passenger Equipment in Steam Railway Service, H. A. Currie and B. F. Wood

Factors in the Selection of Locomotives in Relation to the Economics of Railway Operation, O. S. Beyer, Jr.

Cement Session, 10.00 a.m., Sixth Floor

(Papers contributed by Sub-Committee on Cement Manufacture)

Some Remarks on the Depreciation Factor in the Cost of Producing Portland Cement, Frederick H. Lewis

Discussion by G. S. Brown

The Prevention of Missed Fires in Blasting, W. H. Mason

The Deterioration and Spontaneous Combustion of Gas Coal, Perry Barker

THURSDAY AFTERNOON

Session on Fluid Flow, 2.00 p.m., Sixth Floor

Measurement of Air in Fan Work, Chas. H. Treat

The V-Notch Weir Method of Measurement, D. Robert Yarnall

To be presented by title. Contributed through the courtesy of the Philadelphia Local Committee

The Centrifugal Blower for High Pressures, Henry F. Schmidt
Measurement of Natural Gas, Thos. R. Weymouth

THURSDAY EVENING

Award of the John Fritz Medal for 1912, 8.30 p.m., Auditorium

The John Fritz Medal for 1912 is to be conferred upon Robert Woolston Hunt, Past-President of the Society, "for his contributions to the early development of the Bessemer process." Following the ceremonial will be a joint reception to Mr. Hunt and to the members and ladies of the four American Engineering Societies.

FRIDAY, DECEMBER 6

Administration Session, Auditorium, 10.00 a.m.

Report of Sub-Committee on Administration. The Present State of the Art of Industrial Management

Axioms Concerning Manufacturing Costs, Henry R. Towne
Measuring Efficiency in Manufacturing, Edward B. Passano

To be presented by title. This session if necessary will be continued in the afternoon

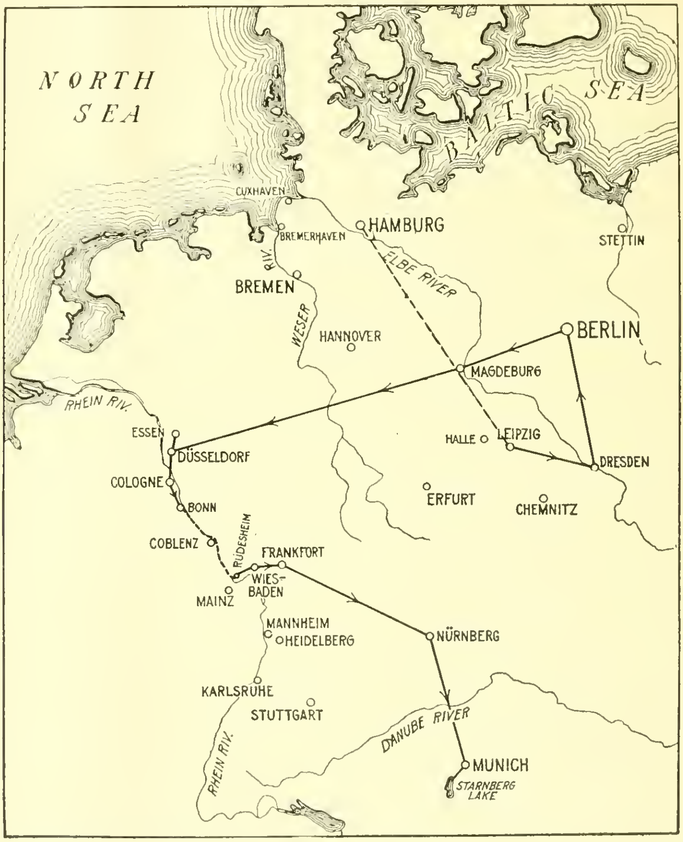
MEETING IN GERMANY

The plans for the meeting of the Society with the Verein deutscher Ingenieure, which promises to be the most remarkable tour of the industries in Germany ever offered to American engineers, continue to go forward actively. Mr. Conrad Matschoss, Dozent of the Royal Polytechnic High School of Berlin who recently visited the United States as a representative of the German society, conferred with the Committee of Arrangements, Col. E. D. Meier, chairman, as to itinerary, professional papers, etc., with most gratifying results.

Owing to the great success of this meeting in which it is probable that many members of the Verein themselves will want to participate, advice has been received by cable that the American party must be limited to 200 members and 100 ladies, since any number in excess of this cannot be accommodated in the hotels of the cities visited. This necessitates the restriction of the party to members only and the ladies of their immediate family. As the number of those who have already signified their intention of joining the party now nearly equals this, any opportunities for participating which may arise must be accorded in the order in which requests are received in the office of the Secretary.

The date of the departure will be June 10 on the Hamburg-American liner Victoria-Luise, arriving in Hamburg on Saturday, June 21. Here the party will be received by the municipal authorities at the point of landing and given an opportunity to visit the great shipyards before going on by special train to Leipzig the same or the following day. Sunday evening, the 22nd, will be the occasion of an informal gathering to promote acquaintanceship, preceding the more formal opening on Monday, June 23. At this opening there will be a welcome by the King of Saxony and the municipal authorities of Leipzig, and two addresses, one by a representative of the German and the other by a representative of the American Society, which will be on general subjects, such as the history of engineering, the relation of capital and labor, and the effect on industry of technical education. Tuesday, the last day to be spent in Leipzig, will be devoted to technical papers and to various social events and participation in the patriotic gathering in commemoration of the one-hundredth anniversary of the battle of Leipzig.

The party will leave on Wednesday for a tour of industrial Germany visiting the principal cities, such as Dresden, Berlin, Düsseldorf, Cologne, Frankfurt a/M., Nürnberg, Munich, etc. In all probability the Krupp works at Essen will be thrown open to the party, the Baroness von Bohlen (geborene von



Itinerary of Official Tour in Germany, June 23-July 7, 1913

Krupp) receiving the visiting American engineers in person. The trip up the Rhine from Bonn to Rüdesheim will be made by boat and at Rüdesheim there will be a grand festival on the embankments. Independence Day will be celebrated at Frankfurt a/M. under the auspices of the American embassy.

The final meeting will be held in Munich, July 7, in connection with the Museum of Technical Arts, of which Dr. von Miller is the Director. It is expected that the Prince of Bavaria and the Mayor of Munich will also receive the party. The entire trip through Germany and the various official receptions are being arranged by a committee of the most important men of the arts and sciences in the German society and there is the keenest interest shown by municipal authorities and the heads of industrial establishments.

The itinerary of the visit will be as follows:

TENTATIVE PROGRAM

June 21 Reception at Hamburg by the local board
June 22 (Sunday) Leipzig, official reception by the Verein deutscher Ingenieure
June 23, 24 Participation in the exercises of the 54th Annual Meeting of the Verein deutscher Ingenieure
June 25, a.m. Dresden
June 27, 28 Berlin
June 29 Düsseldorf. Reception by the Rhine-Westphalian Section of the Verein
June 30 to July 2 Various functions arranged by the Rhine-Westphalian Section, including a trip to Cologne and Essen
July 3 By railway to Bonn. By boat up the Rhine with festival at Rüdesheim. By rail to Frankfurt a/M., arriving about 6 p.m. Evening: reception of welcome at Frankfurt
July 4 Independence Day at Frankfurt
July 5 a.m. Nürnberg
July 6 p.m. Munich. Evening reception of welcome
July 7 a.m. Visit to the German Museum
July 7 p.m. Trip on the Starnberg lake
Final exercises in the Rathhaus

A map illustrating this proposed itinerary is published herewith.

CURRENT AFFAIRS OF THE SOCIETY

The activities of the Society are developing, particularly in the matter of coöperation with other organizations, for the general benefit of the profession, especially along lines of standardization. Members will notice in the report of the Council several committees appointed for specific standardization work, as well as a special committee to advise the Council how we can comprehensively undertake our various duties in this direction.

CONDUCT OF ANNUAL AND LOCAL MEETINGS.

An important and urgent matter before the Society is how we may best conduct our meetings, both general and local, so that members may get the fullest benefit of the Society organization and their interest be maintained, particularly of those who cannot attend the general meetings. The Committee on Constitution and By-Laws have prepared a proposed code, under instruction of the Council, and these Rules and By-Laws, which appear as proposed amendments to the existing By-Laws and Rules, have been issued in pamphlet form to all the standing, geographical and professional committees, with an invitation that comments be sent to the Council. Other members of the Society interested may receive copies of this report upon request. The Council has invited the chairmen of the several committees to attend an informal subscription dinner to be held at the Engineers Club on the evening of Thursday, December 5, for the purpose of discussing these matters fully.

NATIONAL MUSEUM

Some of our most active members have long desired to see the establishment of a Museum where the development of engineering and the arts might be preserved, and in the original design of the Engineering Societies Building a large room on the twelfth floor was set apart for this purpose, which on account of lack of funds has never been accomplished. The visit of Dr. Oskar von Miller and the commission from Munich of the German Museum, however, has revived this interest with such enthusiasm that the Council has appointed a committee to coöperate with other organizations in the establishment of a great National Museum.

1915 ENGINEERING CONGRESS

The Society is more sensitive than ever to the interests of all of its members and takes great satisfaction in the prospect of an International Engineering Congress in San Francisco in 1915 to celebrate the world's greatest engineering achievement, the Panama Canal. As will be seen in the minutes of its last meeting, the Council has been pleased to underwrite this congress to the extent of $5000, or any portion of it which may be its share of the expenses, provided the other American engineering societies similarly participate.

JOHN FRITZ MEDAL

Invitations have been mailed to the entire membership, notifying them of the award of the John Fritz Medal to a Past-President of this Society. Mr. Robert Woolston Hunt. As the award will be made on Thursday evening during our Annual Convention, the Council has decided to omit the annual reunion, as it will on this occasion act as joint host with the American Institute of Electrical Engineers and the American Institute of Mining Engineers to all the members and ladies of the four American engineering societies.

Calvin W. Rice, *Secretary.*

AMENDMENTS TO BY-LAWS AND RULES

AMENDMENT TO BY-LAW 18

B 18 The Council at any meeting may, in its discretion, permanently remit the dues of any Member of the Society who has paid thirty years' dues, or who shall have reached the age of seventy years after having paid twenty-five years' dues, provided that notice of such proposed action shall have been given at a previous meeting of the Council and the Committee on Membership shall have concurred in recommending that this action be taken. The Council may, in its discretion, restore to membership any person dropped from the rolls for non-payment of dues, or otherwise, upon such terms and conditions as it may at the time deem best for the interests of the Society.

This announcement appeared in the November issue, but with an error in the wording of the By-Law,

AMENDMENTS TO RULES

R 13 The features of the program outside of the professional and business sessions to be provided for the members and guests at any meeting of this Society in any city shall be in charge of a Local Committee, subject, however, to the general approval of the Committee on Meetings. The President's reception of the annual meeting shall be in charge of the House Committee under the general supervision of the Committee on Meetings.

PROFESSIONAL SECTIONS AND GEOGRAPHICAL GROUPS

R 19 A professional section or a geographical group of the Society shall consists of Honorary Members, Members, Associates, and Juniors of The American Society of Mechanical Engineers and of other persons to be designated Affiliates as hereinafter described.

R 20 A professional section of The American Society of Mechanical Engineers may, with the approval of the Council, be organized for the consideration of any engineering, scientific, or professional topic, provided that a number satisfactory to the Council, of members of The American Society of Mechanical Engineers, unite in making written request for such an organization. A similar procedure shall apply to geographical groups. Such a section or group shall be designated as —— Section or —— Group of The American Society of Mechanical Engineers,—the blank being filled by the topic specialized or name of locality.

R 21 The provisions of the Constitution, By-Laws and Rules of The American Society of Mechanical Engineers, and the precedents of the Society with respect to professional sessions for the discussion of papers, shall cover the procedure of the professional sections or geographical groups except that no meeting of a section or group shall be considered a meeting of the Society as a whole.

R 22 For the convenient conduct of its professional affairs the section or group shall organize an Executive Committee of five members of the Society, under the general direction of the Council. Such officers as the section or group shall require must be selected from the membership of the Society. Other committees of the section or group shall be appointed by its Executive Committee.

R 23 The Executive Committee of the section or group, subject to the approval and direction of the Secretary of the Society, shall designate a Secretary of the section or group, whose duties shall be those usually attaching to the Secretary of a professional session, and who shall also see that the discussions of papers are satisfactorily reported and transmitted to the Secretary of the Society.

R 24 Expenditures for the purpose of a section or group chargeable to the Society must be authorized by the Secretary of the Society before they are incurred, and must be provided for in the estimate and budget approved by the Council. No liability otherwise incurred shall be binding on the Society. Any expenditure not so provided must be met by the section or group itself.

R 25 Engineers and others not members of The American Society, but desiring to participate in the meeting of the section or group, may enroll themselves as affiliates as heretofore provided, with the approval of the Executive Committee of the section or group. Such affiliates shall have the privilege of presenting papers and taking part in the discussions. They shall pay $5 per annum, which shall be due and payable in advance, on October 1st of each year of their enrollment, and shall thereby be entitled to receive the regular issues of The Journal for a period covered by their dues.

R 26 The Council of The American Society of Mechanical Engineers may, at sixty days notice, suspend or disband any section or group.

COUNCIL MEETING, NOVEMBER 12

At a meeting of the Council held on Tuesday afternoon, November 12, there were present Alex. C. Humphreys, E. D. Meier, James Hartness, H. G. Reist, Henry Hess, Jesse M. Smith, Ira N. Hollis, F. R. Hutton, E. B. Katte, Wm. H. Wiley, George A. Orrok, R. M. Dixon, and C. W. Rice, Secretary. The following deaths were reported: R. P. Badeau, F. H. Crabtree, A. W. Danforth, S. T. Mudge, J. M. Robinson, A. E. Rowland, George H. Schuete, J. B. Siner and Maunsel White. Resignations were accepted from Percy Allan, C. W. Baker, G. H. Benton, A. S. Blanchard, H. M. Chamberlain, E. D. Church, H. W. T. Collins, Theo. F. Dreyfus, J. H. King, G. P. Lawrence, W. G. Marot, Harris J. Ryan, C. M. Russell, Harry Tandy, Jas. C. Porter, F. H. Burton, A. T. Doud, John R. James, W. A. Schaefer.

Fred J. Miller, chairman, R. M. Dixon, Morris L. Cooke and Carl J. Barth were appointed a Committee on Codification of Office Standards.

Amendments to the Rules, in respect to the conduct of professional sessions and geographical groups, which appear elsewhere in this issue, were passed.

H. G. Stott, A. F. Ganz and Carl Schwartz were appointed a committee to coöperate with the American Institute of Electrical Engineers in the matter of standardization of the new unit of power, the myrowatt.

A committee consisting of A. L. DeLeeuw, Henry Hess and John Riddell were appointed to coöperate with the National Machine Tool Builders Association in the matter of standardization of machine tools.

C. J. Davidson, Newell Sanders, Putnam A. Bates and Wm. Butterworth, were appointed a committee to coöperate with the

Society of Agricultural Engineers in the standardization of agricultural implements.

A committee composed of E. D. Meier, H. G. Reist, Ambrose Swasey, George Mesta, and Geo. F. Kunz was appointed to coöperate with other organizations in the establishment of a National Museum.

A committee consisting of Henry Hess, J. H. Barr, Charles Day, C. J. Davidson and Carl Schwartz, was appointed to advise the Council on the general conduct of the standardization work of the Society.

In connection with the International Congress to be held in San Francisco in 1915, $5000 or such portion of it as may be our share, was ordered underwritten, provided the other American societies make similar provision for this congress.

The Council was very gratified to receive an inquiry from the Milwaukee Engineering Society with respect to affiliation with this Society, and appointed Geo. M. Brill, Vice-President, to extend the greetings of the Society to the Milwaukee organization.

The Secretary was authorized to appoint H. A. Hey assistant to the Secretary, to represent the Secretary in his absence.

The Annual Reports of the Council and Standing Committees were received and referred jointly to the Executive and Publication Committees.

The meeting adjourned to Tuesday, December 3, at two o'clock.

LAND AND BUILDING FUND

Through the willingness of the membership to assume the responsibility of our indebtedness on the land fund of the Engineering Societies Building, our share of this amount has now been paid off, so that the Society may feel itself free to undertake the new activities and responsibilities which confront it.

At a meeting of the Council on June 18, the following action was taken:

Whereas direct official acknowledgement has not been made of all the contributions to the Land and Building Fund of the Society and to owners of Certificates of Indebtedness, and only a partial list printed in The Journal, be it

Resolved: That an alphabetical list of all contributors to the fund and owners of certificates be printed in an early issue of The Journal, and be it

Resolved: That the thanks of the Society are hereby tendered to the contributors to the Land and Building Fund and to the owners of Certificates of Indebtedness, by which The American Society of Mechanical Engineers has been able to discharge its obligations in connection with its participation in the gift by Mr. Andrew Carnegie, Honorary Member of The American Society of Mechanical Engineers of the Engineering Societies Building.

Resolved: That the thanks of the Society are hereby tendered to the members of the Land and Building Fund Committees, Messrs. Fred J. Miller, Chairman, James M. Dodge, R. C. McKinney, Committee in 1907; Worcester R. Warner, Chairman, F. H. Stillman (deceased), George A. Orrok, I. E. Moultrop and C. N. Lauer, Committee in 1910; and R. M. Dixon, Chairman, Waldo H. Marshall and Charles Wallace Hunt (deceased), Committee in 1911, with respect to the Certificates of Indebtedness, for the able and energetic manner in which these committees conducted the work intrusted to them, which resulted in securing contributions from 66 firms and 300 members and in placing 163 certificates of indebtedness, aggregating $161,000.

Resolved: That copy of these resolutions be sent to the members of each of the above committees.

Resolved: That the Society is under special obligation to Charles Wallace Hunt, Past-President, for his wisdom and foresight, after exhausting the opportunities for further contributions, in providing for the discharge of the remaining obligations of the Society, $81,000 by the issue of Certificates of Indebtedness and for the preparation of the presentation letter with respect thereto, which resulted in an over-subscription.

The list of names is as follows:

CONTRIBUTORS TO LAND FUND

Abbott, W. L.
Abercrombie, J. H.
Alden, G. L.
Allen, W. M.
Allis-Chalmers Co.
Almond, T. R. Mfg. Co.
American Locomotive Co.
American Pulley Co.
Andrews, Jas. D.
Appleton, Wm. D.
Babcock & Wilcox Co.
Baird Machinery Co.
Baker, J. C. W.
Baldwin, Stephen
Bancroft, J. Sellers
Barnes, S. G.
Barnes, W. F.
Barth, Carl G.
Bartlett, Henry
Bates, A. H.
Bates, Edward P.
Bausch, Wm. H.
Baylis, Arthur R.
Beck, Jas. D.
Bettendorf, W. P.
Bickford Drill & Tool Co.
Bilgram, Hugo
Billings, C. E.
Billings & Spencer Co.
Bitterlich, Walter J.
Blauvelt, A.
Borden, W. H.
Bradford Mch. Tool Co.
Brandon, G. R.
Breckenridge, L. P.
Bridgeport Mfg. Co.
Brill, Geo. M.
Brown & Sharpe Mfg. Co.
Brown, A. E.
Brown, A. T.
Brownell Company,
Brush, C. F.
Buckeye Engine Co.
Buckley, J. F.
Bullard Machine Tool Co.
Bump, B. N.
Burlingame, L. D.
Burns, A. L.
Burroughs Adding Mch. Co.
Burton, J. H.
Cadillac Motor Car Company
Calder, John
Caracristi, V. Z.
Carlsson, C. A. V.
Cheney, W. L.
Christie, W. W.
Cinn. Machine Tool Co.
Cinn. Milling Machine Co.
Cinn. Plane Co.
Clegg, Robert I.
Clemens, Chas. W.
Cogswell, W. B.
Coleman, E. P.
Colts Patent Firearms Mfg. Co.
Conard, W. R.
Continental Iron Works
Cooke, Harte
Core, W. Wallace
Cox, J. D.
Cromwell, J. C
Doty, Paul
Dodge, Day & Zimmermann
Dougherty, S. B.
Dow, Alex.
Dreses Machine & Tool Co.
Duncan, J. D. E.
Dunn, J. W.
Dyer, R. A., Jr.
Eberhardt, Elmer G.
Eberhardt, Frank E.
Eberhardt, Henry E.
Eberhardt, Henry J.
Edson, J. B.
Edwards, V. E.
Ekstrand, C.
Ellsworth, J. W. & Co.
Engineering Magazine
Engineering News

Engineering Record
Evans, H. O.
Falkenau, A.
Ferguson, J. W.
Ferracute Machine Co.
Ferry, C. H.
Fish, C. H.
Fleming, H. S.
Fletcher, W. & A. Co.
Flinn, Thos. C.
Flint, Walter
Foote, A. W.
Freeman, John R.
Freeman, S. E.
French, E. V.
Fritz, John
Fuchs, Ernesto
Gabriel, Wm. A.
Gaehr, David
Gamper, H.
Gardiner, A. J., Jr.
Gately, P. J.
Gates, P. H.
Gisholt Machine Co.
Glasgow, A. G.
Gleason Works
Goss, W. F. M.
Green, Wm. O.
Guelbaum, David
Guckel, Chas. H.
Harrington, N. T.
Hartness, Jas.
Haskell, B.
Hayes Mfg. Co.
Hazard, F. R.
Heaton, H. C.
Hecker, H. A.
Heine Safety Boiler Co.
Helmes, M. J.
Henderson, Richard
Henry, George J., Jr.
Herbert, J. S.
Herbert, C. G.
Hess-Bright Mfg. Co.
Hess, Henry
Hubbard, H. D.
Higgins, M. P.
Hill, H. H.
Hill Publishing Co
Hinchman, C. R.
Holcomb, A. E.
Holloway, H. F.
Hosmer, Sydney
Houston, Stanwood & Gamble Co.
Howard, C. P.
Hulett, F. E.
Hulett, G. H.
Hunt, Chas. W.
Hunter, Geo. E.
Huson, W. S.
Hutchinson, Richard
Hutton, F. R.
Idell, Frank E.
Industrial Press
Isham, Henry S.
Jackson, D. C.
Jacobus, D. T.
Jenkins Brothers
Jenks, R. P.
Johnson, C. W.
Johnston, F. W.
Jones, Horace K.
Katte, E. B.
Kelley, Fred W.
Kennedy, F. L.
Kennedy, M. E.
Kennedy Valve Mfg. Co.
Kent, J. M.
Kimber, G. A.
Kirby, Frank E.
Kirchhoff, Chas.
Kwong Yung Kwang
Laird, W. G.
Lane, H. M.
Lange, P.
Lardner, Henry A.
Larner, C. W.
Larsson, T. L. F.
Lauder, G. B.
Lauer, C. N.
LeBlond, R. K., Mch. Tool Co.
Leland, Henry M
Leonard, E. J.
Lewis, Wilfred
Lidgerwood Mfg. Co.
Link Belt Co.

Little, Chas. H.
Littleford Bros.
Lobben, Peder
Locomobile Co. of America
Lodge & Shipley Mch. Co.
Loring, Chas. H.
Lucas Machine Tool Co
Lunkenheimer Company
Lyall, Wm. L.
MacArthur, Robt. J.
McIntosh, Seymour & Co.
McMillen, Emerson
Machold & Riddell
Magruder, Wm. T.
Main, Chas. T.
Malvern, L. K.
Manning, C. H.
Martell, L. H.
Martin, F. S.
Marx, Henry
Mason, Wm.
Mason, Wm. B.
Mattice, A. M.
Maury, D. H.
Meaker, Guy L.
Meier, E. D.
Melville, George W.
Merryweather, G. E.
Messnick Mfg. Co.
Miller, Fred J.
Miller, Theo. H.
Miller, L. B.
Mix, E. W.
Moeller, F.
Moore, Chas. C. & Co.
Morgan, R. L.
Morrin, Thos.
Morrison, H. S.
Moultrop, I. E.
Mount, W. D.
Murphy, B. S.
Murphy, John K.
Murphy, John Z.
Naegeley, John C.
Nate, E. H.
Neave, P. M.
Newark Gear Cutting Machine Co.
Niles Bement Pond Company
Norris, R. V.
Otis, Spencer
Park, W. R.
Parkhurst, F. A.
Pettis, C. D.
Philbrick, Frank B.
Platt, John
Polson, J. A.
Power, Fred M.
Prather, H. B.
Pratt, Francis C.
Prescott, F. M.
Price, A. M.
Prosser, Jos. G.
Quimby, W. E.
Quint, A. D.
Quintard Iron Works
Reed, E. Howard
Reid, Marcellus
Reiss, George T.
Reist, H. G.
Rhoades, G. E.
Rice, Calvin W., Sec'y
Richter, Ernest
Rider, George S.
Riddell, John
Riggs, John D.
Rivett, E.
Robertson, R. A.
Robeson, A. M.
Roe Stephens Mfg. Co.
Roper, Norman B.
Sargent, Fred'k
Satherberg, Carl H.
Saunders, W. O.
Saxon, Wm
Schaeffer, Jos. C
Scheele, John M. E.
Scheffler, Fred A
Schneider, Paul E.
Schulte, G. H.
Schwarz, F. H.
Sellers, Wm. & Co., Inc
Sharpe, H. D.
Shirrell, D.
Smith, E. J.
Smith, Jesse M.
Smith, Oberlin

Smith, O. G.
Snow, Wm. G.
Snow, Wm. W.
Solvay Process Co.
Sprado, Ralph
Stanwood, J. B.
Starbuck, G. F.
Stetson, G. R.
Stevens, John A.
Stut, J. C. H.
Sulzer, J. Carl
Swasey, Ambrose
Sweet, John E.
Symonds, N. G.
Terry, C. D.
Thomas, Charles W.
Thompson, A. W.
Thompson, John
Tower, D. W.
Tompkins, Stonewall
Torrance, Henry
Towle, Wm. M.
Towne, Henry R.
Triumph Electric Co.
Trump, Edw. N.
Tufts, Leonard
Turner, Charles P.
Turner, John
Tuttle, W. B.
Tyler, Chas. C.
Underwood Typewriter Company
Union Twist Drill Company
United Engrg. & Fdy. Co.
Upson, Maxwell M.
Vauclain, S. M.
Vaughan, H. H.
Veeder, C. H.
Viola, B.
Waitt, A. M.
Wallace, D. A.
Ward, Charles E.
Wards Engine Works
Warren, John E.
Warner, W. R.
Watson-Stillman Company
Weber, H. F.
Wehner, Lewis
Wellman, S. T.
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Weymouth, C. R.
Wheeler, C. H.
Whitaker, H. E.
White, John C.
Whitehead, George E.
Whitney Mfg. Co.
Whyte, John S.
Wicks, Henry
Wildin, G. W.
Wiley, John & Sons
Willcox, Chas. H.
Williamson, George E.
Willis, E. J.
Wolf, Fred W.
Wood, Erwin E.
Wood, Walter
Woodbury, C. J. H.
Young, John P.

HOLDERS OF CERTIFICATES OF INDEBTEDNESS

Abbott, W. L.
Abercrombie, J. H.
Alden, G. I.
Alexander, M. W.
Allen, Francis B.
Allison, Robt.
Anderson, J. M.
Appleton, C. B.
Ayers, H. B.
Bailey, H. H.
Baldwin, C. Kemble
Bancroft, J. Sellers
Barr, John H.
Barth, Carl G.
Basford, G. M.
Best, W. J.
Bond, Geo. M.
Bourne, Geo. L.
Bower, E. S.
Brill, Geo. M.
Caracristi, Virginius Z.
Carroll, Elbert H.
Carpenter, J. M.
Chester, J. N.

Christie, W. W.
Clark, F. H.
Cobb, Geo. H.
Colwell, A. W.
Colwell, J. V. V.
Conlee, Geo. D.
Conrad, W. W.
Cooke, Fred. W.
Coster, Edward L.
Daniels, F. H.
Deane, F. W.
Dickerman, W. C.
Dixon, R. M.
Doane, W. H.
Doelling, L. K.
DuPont, L. C.
Durant, W. F.
Dyer, R. A., Jr.
Eaton, H. C.
Ely, Theo. N.
Emerson, Harrington
Emmet, W. L. R.
Estes, Wm. W.
Farquhar, Francis
Fermier, E. J.
Firmstone, F.
Flagg, Saml. B.
Flory, B. P.
Foran, Geo. J.
Ford, Bruce
Foster, C. H.
Fraser, D. Ross
Fraser, Norman D.
Fritz, John
Gantt, H. L.
Garden, H. C.
Gazzam, Joseph P.
Geier, Fred. A.
George, J. T.
Gordon, H. D.
Goss, W. F. M.
Grossmann, A.
Hadfield, R. A.
Hammer, Alfred E.
Hammett, H. G.
Hartness, Jas.
Herbert, C. G.
Herr, E. M.
Herschel, W. A.
Hess, Henry
Hinchman, C. R.
Hoefer, E. J.
Hoffman, J. D.
Holmes, Arthur
Honiss, Wm. H.
Honsberg, August A.
Hoxie, W. D.
Hugo, T. W.
Humphreys, Alex. C.
Hunt, Wm. F.
Hussey, W. E.
Hutchins, Geo. F.
Hutton, F. R.
Jackson, Arthur C.
Jacobi, Albert W.
Jenks, R. P.
Johnstone, F. W.
Jones, H. C.
Libby, Malcolm
Lindstrom, Chas. A.
Longstreth, Chas.
Lucas, H. M.
Lundgaard, I.
Main, Chas. T.
Manning, C. H.
Marks, Lionel S.
Marshall, W. H.
Matheson, W. G.
Meier, E. D.
Melville, G. W.
Merz, Robt. G.
Miller, Fred J.
Morgan, Paul B.
Moultrop, I. E.
Muhlfeld, J. E.
Munson, E. G.
Murphy, Edw. J.
McBride, James
McDewell, H. S.
Nelson, Jas. W.
Nickel, F. F.
Nickerson, Chas. H.
Nickerson, A. T.
Otis, Spencer
Phetteplace, T. M.
Plunkett, Chas. T.

Posey, Jas.
Pratt, Jas. A.
Reid, John S., Sr.
Reiss, Geo. T.
Richter, Ernest
Riddell, John
Ritter, H.
Robinson, A. W.
Rohlig, George G.
Ray, David H.
Royle, Vernon
Sague, J. E.
Sanguinetti, Philip C.
Sando, W. J.
Schwamb, Peter
Schwartz, Carl
Sellers, Coleman, Jr.
Sharpe, Henry D.
Shedd, Frank E.
Small, H. J.
Smith, Augustus
Smith, Harry J.
Smith, Jesse M.
Smith, J. W.
Spalding, C. M.
Spiro, Chas.
Stearns, A. W.
Stafford, B. E. D.
Stetson, G. R.
Stott, H. G.
Stutz, C. C.
Swain, Geo. F.
Swasey, Ambrose
Sweet, J. E.
Taylor, F. W.
Taylor, J. W.
Thompson, O. C.
Towne, H. R.
Tyler, C. C.
Vauclain, S. M.
Viall, W. A.
Wadleigh, Geo. R.
Waitt, Arthur M.
Warren, John E.
Westinghouse, H. H.
Wiley, Chas.
Wiley, Wm. H.
Whyte, F. M.
Zehnder, C. H.

ANNUAL REPORT OF COUNCIL AND STANDING COMMITTEES

REPORT OF THE COUNCIL

The activities of the Society for the year 1911-1912 have covered a wide scope. Prominent among them has been coöperation with other societies. For this purpose a conference committee has been formed, consisting of two appointees each from the American Society of Civil Engineers, the American Institute of Mining Engineers, the American Institute of Electrical Engineers and The American Society of Mechanical Engineers, whose province it is to bring to the attention of their respective organizations for action matters of importance. The Society has also coöperated with other organizations and appointed committees to assist.

A Committee on Engineering Standards has been formed, with which the prominent engineering societies of the world have been asked to coöperate, to establish a clearing house for the issuance of information respecting standards for the engineering profession.

At various functions during the year, the Society has been represented by the following Honorary Vice-Presidents, appointed by the Council: William M. Rice Institute, Houston, Texas, W. B. Tuttle; Fourth National Conservation Congress, George M. Brill and W. B. Jackson; American Electric Railway Association, R. W. Hunt and C. J. Davidson; International Rubber Conference, E. B. Katte; International Conference on Hygiene and Demography, H. de B. Parsons and T. Kennard Thompson; Pratt Institute Commencement Exercises, C. W. Obert; Carnegie Institute of Technology, O. P. Hood; National Waterways Commission, H. G. Stott; International Congress of Navigation, Wm. T. Donnelly; American Academy of Political and Social Science, Geo. W. Melville, C. M. Day, J. M. Rusby.

Ties with organizations abroad as well as at home are continually being strengthened by the Society's policy of coöperation. During the year the Society has participated in the erection of a memorial window at Westminster Abbey to the memory of Lord Kelvin, under the auspices of the Institute of Civil Engineers, $600 having been subscribed by members of the Society for this purpose. In April of this year, the members of the German Museum Commission appointed to study important engineering achievements in America, were given the greetings of the Society at a meeting in the Engineering Societies Building, and in the various cities visited were assisted by committees of the membership formed for the purpose. An invitation to participate in a meeting of the Verein deutscher Ingenieure in Leipzig in the summer of 1913 has been accepted by the Council and active preparations for the trip have been begun by the Committee of Arrangements, Col. E. D. Meier, chairman.

Within the Society itself new activities have been undertaken. As a result of an earnest consideration by the Council of ways and means of extending the membership, a committee on Increase of Membership, I. E. Moultrop, chairman, was appointed early in the year. This committee has coöperated with the Membership Committee in seeking to invite only those qualified for admission from among those prominent in the profession and if possible increasing rather than lowering in any way the present high standard of membership. The proposed amendments to the Constitution provide an additional engineering grade, making the several grades of membership similar to the more important American and English societies and simplifying the method of election in accordance with these same prominent organizations. The proposed engineering grade is that of Associate-Member, intermediate between Member and Junior, and election by the Council is proposed to be substituted for election by the membership.

Honorary membership has been conferred during the year upon Anatole Mallet of Paris. Carl Gustav Patrick DeLaval, Stockholm, Rudolph Diesel, Munich, and Oskar von Miller, Munich. The death of George Wallace Melville, Past-President and Honorary Member, is reported with regret.

The following table shows the changes in membership for the fiscal year, October 1, 1911, to October 1, 1912, in distinction from the administration year which the rest of the report covers:

MEMBERSHIP FOR FISCAL YEAR (OCT. 1, 1911–OCT. 1, 1912)

Grade	Oct. 1, 1911	Losses				Additions			Net decrease	Net increase	Oct. 1, 1912
		Transfer	Resignations	Lapsed	Death	Transfer	Election	Reinstatements			
Honorary......	16	..	..	..	1	..	3	..	..	2	18
Members......	2702	1	19	30	27	54	278	4	..	259	2961
Associates.....	370	13	4	10	4	2	16	..	13	..	357
Juniors........	919	43	17	28	2	1	175	1	..	87	1006
Total.....	4007	57	40	68	34	57	472	5	..	335	4342
Affiliates Gas Power Section.	134	....				....			8	..	126
Affiliates Student Section..	635	....				....			..	87	722

The University of California has been added to the list of student branches, which now number 25. The total number of student affiliates is now 722. The Committee on Student Branches has been increased by the addition of Wm. Kent and George A. Orrok, with the intent that through this enlarged committee visits can be made to the several branches.

An important step with reference to local meetings of the Society was taken at the Spring Meeting in Cleveland, after conference with representatives of the several cities where meetings are held. The policy adopted was to extend the rules for professional sections making them applicable to geographical sections. Upon approval of the Council local meetings may now be considered sectional meetings, with the local committee in responsible charge. A close relation between the Society's headquarters and those of the several centers is maintained by the interchange of papers. Under these provisions meetings as sections are now being conducted in San Francisco, St. Louis and Cincinnati.

The Council reports the final adoption of amendments to the Constitution, By-Laws and Rules, which appear elsewhere.

Committee activity has been marked throughout the year and the advance which the Society is making has come about through the earnest work of the standing and special committees. Full details appear in the separate reports of the committees accompanying this report.

The task of completing the series of pictures of Honorary Members has progressed under the care of the House Committee, and twenty-two of these portraits are now hung upon the walls of the Society's rooms. To this committee has been assigned the duties in connection with the President's Reception given each year at the Annual Meeting.

Through the initiative of the Library Committee amendments to the By-Laws of the United Engineering Society have been prepared and have received the approval of the Boards of each of the Societies comprising the United Engineering Society, whereby a common administration of the joint libraries of the American Institute of Mining Engineers, the American Institute of Electrical Engineers and The American Society of Mechanical Engineers, may be secured, making possible greater efficiency in the joint library service, together with symmetry of the several collections, so that there may be no duplication of effort.

An important advance movement has been undertaken by the Committee on Meetings in the appointment of six additional sub-committees, covering various phases of mechanical engineering, with the idea of securing increased service through the assistance of specialists who will report developments in their respective fields. There are now ten of these sub-committees, covering Administration, Air Machinery, Cement Manufacture, Fire Protection, Hoisting and Conveying, Industrial Building, Iron and Steel, Machine Shop Practice, Railroads, and Textiles.

Due to the rapid growth of the Society, the work of the Publication Committee has become increasingly difficult, particularly as regards Transactions, there being sufficient important material to fill a two volume edition. The Council has concurred in the recommendation of the committee that an index to Transactions, comprising volumes 1 to 33 inclusive, be published, and an appropriation has been secured for the purpose. The subscription price of The Journal has been reduced to $2 a year to members and affiliates and $3 a year to non-members. This publication has been entirely self-supporting during the year and has continually sought to enlarge its field of activity and usefulness, so as to present a complete record of mechanical engineering.

A total of 865 applications for membership have been received by the Membership Committee during the fiscal year, of which 694 have been recommended for membership. Eight former members have also been reinstated. The Committee have adopted a more simple form of application and reply blank which will more readily meet its requirements.

A special committee on Code of Ethics has been appointed by the Council, following the example of other societies, to consider the formation of such a code.

The special committee on Flanges has coöperated with the National Association of Master Steam and Hot Water Fitters in formulating the 1912 schedule of standard weight and extra-heavy flanges and flanged fittings. This has been adopted by the Government in several of its departments and by numerous organizations.

As a result of interest awakened in the subject of measurement of taps, bolts and screws, a committee has been appointed to take up the matter, whose duty it shall be to prescribe the permissible tolerances in the commercial manufacture of these products, including their measurement.

The committee on Standard Cross-Sections and Symbols will report at the Annual Meeting on standards for drawings to indicate the materials used.

Upon invitation of the Patent Law Association, a conference was held in Washington at which the Society was represented, together with several other national societies. The Committee on Patent Law has been continued so that it may attend any conferences which may arise and participate in any developments, reporting to the Society as the necessity may arise.

At the Annual Meeting the very complete and remarkable report of the Committee on Power Tests will be presented and thus bring up to date and extend the present reports of the Society on these subjects.

REPORT OF FINANCE COMMITTEE

Appended will be found the certified report of the examination of the accounts of the Society, made by Messrs. Peirce, Struss & Company, for the fiscal year of the Society, ending September 30, 1912.

Your Finance Committee reports that the total income of the Society for the fiscal year ending September 30, 1912, was $112,535.06, being an increase of $17,453.32 over the previous year. These increases are largely due to the very excellent work of th Increase of Membership Committee, the Sales Department and the Advertising Department.

The total expenditures of the Society were $101,306.25, an increase of $6,915.42.

Your Committee feels that it must not neglect to call attention to the great expenses charged to the account of the Membership Committee, which amounts to over $5 for each new member joining the Society. This expense all accrues after the application is received and includes the cost of the Certificate of Membership, about $1 apiece. Your Committee respectfully urges that the Membership Committee gives especial attention to this matter with a view to eliminating a large proportion of this expense.

The Publication Committee has spent but $44,516.40, out of an appropriation of $48,300, but there are some expenses, estimated to amount to $4200, in connection with Volume 33 of Transactions, which have already accrued but are not paid. The Finance Committee recommends that this amount be set aside from the income of the year ending September 30, 1912, for meeting this expense.

It is estimated that the income of the Society for the year ending September 30, 1913, will be $117,329, and your Finance Committee recommends the following appropriations, amounting to $106,775:

Finance Committee	$23,375
Membership Committee	2,500
Increase of Membership Committee	1,800
House Committee	2,000
Library Committee	4,500
Meetings Committee	5,500
Council	7,000
Publication Committee	50,500
Power Tests Committee	1,000
Research Committee	100
Society History Committee	500
Sales Expenditures	8,000
	$106,775

Your Committee would further advise that as the year progresses close watch be kept of the income and expenses and that if the income should increase or decrease the appropriations should be modified accordingly, being maintained in the aggregate at about 90 per cent of the expected income.

Respectfully submitted,

R. M. Dixon, *Chairman* W. H. Marshall H. L. Doherty W. L. Saunders W. D. Sargent	*Finance Committee*

APPENDIX TO REPORT

November 1, 1912.

Mr. R. M. Dixon,
Chairman Finance Committee

Dear Sir:

In accordance with your instructions, we have audited the books and accounts of The American Society of Mechanical Engineers for the year ended September 30, 1912.

The results of this examination are presented in three exhibits, attached hereto, as follows:

Exhibit A Balance Sheet, September 30, 1912

Exhibit B Income and Expenses for the year ended September 30, 1912

Exhibit C Receipts and Disbursements for the year ended September 30, 1912

We beg to present, attached hereto, our Certificate to the aforesaid exhibits.

Peirce, Struss & Co.,
Certified Public Accountants

November 1, 1912

Mr. R. M. Dixon.
Chairman Finance Committee

Dear Sir:

Having audited the books and accounts of The American Society of Mechanical Engineers for the year ended September 30, 1912, we hereby certify that the accompanying Balance Sheet is a true exhibit of its financial condition as of September 30, 1912, and that the attached statements of Income and Expenses, and Cash Receipts and Disbursements are correct.

Peirce, Struss & Co.,
Certified Public Accountants

EXHIBIT A

BALANCE SHEET, SEPTEMBER 30, 1912

ASSETS

Equity in Society's Building (25 to 33 West 39th Street)	$353,346.62	
Equity one-third cost of land (25 to 33 West 39th Street	180,000.00	
		$533,346.62
Library Books	13,000.00	
Furniture and Fixtures	5,000.00	
		18,000.00
New York State 3½% Bonds 1954 (Par $35,000.00)	30,925.00	
Cash in Banks representing Trust Funds	11,320.78	
		42,245.78
Stores including Plates and finished publications		13,076.70
Cash in Banks for general purposes	21,322.32	
Petty Cash on hand	350.00	
		21,672.32
Accounts Receivable		
Membership Dues	8,635.75	
Initiation Fees	1,155.00	
Sale of Publications, Advertising, etc	10,400.69	
		20,191.44
Bills receivable		612.45
Advance payments		1,675.54
Total Assets		$650,820.85

LIABILITIES

Certificates of Indebtedness		$74,800.00
Funds		
Life Membership Fund	$35,386.07	
Library Development Fund	4,902.71	
Weeks Legacy Fund	1,957.00	
		42,245.78
Dues paid in advance		1,559.13
Initiation Fees uncollected		1,155.00
Unappropriated Revenue		11,228.81
Reserve (Initiation Fees)		35,759.89
Surplus in Property and Accounts Receivable		484,072.24
Total Liabilities and Surplus		$650,820.85

EXHIBIT B

Income and Expenses for the Year Ended September 30, 1912

INCOME

Membership Dues	$61,198.01	
Sales, Gross Receipts	10,908.15	
Advertising	38,562.22	
Interest and Discount	1,866.68	
Miscellaneous	......	
Total		$112,535.06

EXPENSES

Finance Committee	$25,889.43	
Membership Committee	3,572.04	
Increase of Membership Committee	1,887.53	
House Committee	1,895.31	
Library Committee	1,521.64	
Meetings Committee	6,083.69	
Council Committee	4,438.09	
Publication Committee	44,516.40	
Research Committee	101.75	
Power Tests Committee	50.00	
Sales Expenditures	7,750.37	
Reserve Fund	600.00	
Total		$101,306.25
Excess of Income over Expenses		$11,228.81*

*$4200 of this amount was reserved to complete Trans., vol. 33.

EXHIBIT C

Receipts and Disbursements for Year Ended September 30, 1912

RECEIPTS

Membership Dues	$59,270.81	
Initiation Fees	10,270.00	
Membership Dues and Initiation Fees paid in advance	1,485.80	
Sales of publications, badges, advertising, etc.	48,155.94	
Subscription to Land Fund	344.00	
Interest	2,479.92	
Certificates of Indebtedness	35,149.04	
	$157,155.51	
Cash in Bank and on hand, September 30, 1911	31,081.10	
		$188,236.61

DISBURSEMENTS

Disbursements for general purposes	$114,243.51	
Cancellation of Mortgage	41,000.00	
Cash in Banks and on hand, September 30, 1912	32,993.10	
		$188,236.61

REPORT OF THE HOUSE COMMITTEE

The work of the House Committee during the past year has been generally confined to routine matters. No further material additions have been made to the decorations or furnishings of the rooms of the Society. Some improvements in ventilation of the rooms and offices have been effected and new equipment purchased to facilitate the business work of the Society. A number of additional portraits of honorary members have been secured and framed, and present prospects indicate the substantial completion of this collection in about two years' time.

Respectfully submitted,

FRANCIS BLOSSOM, *Chairman*
EDWARD VAN WINKLE
H. R. COBLEIGH
S. D. COLLETT
W. N. DICKINSON
House Committee

REPORT OF THE COMMITTEE ON MEETINGS

With reference to financial expenditures, reading and judging of papers, the number of meetings held, and similar routine matters, the work of the Committee on Meetings has been for the past year substantially on the same basis as for the preceding year. At no time in the year is any member of this committee free from Society service and the continuity of this demand cannot be realized by any one who has not served on it. Most of the work is conducted by correspondence, but in no case is any committee action taken without substantially unanimous approval. Through the winter months meetings are held weekly or bi-weekly, as the business demands, for the clearing up of matters that cannot be disposed of by correspondence and for verbal discussion on questions of policy.

The service that the Society demands of the Committee on Meetings is considerably more in quantity and quality than a single committee of five men can efficiently perform. It is this feeling that prompted the recommendations last year for the effective enlistment of the Society membership in assisting the work of the Committee on Meetings by organizing sub-committees. It is, therefore, with considerable satisfaction that your committee is able to report not only the inauguration of this plan but its successful operation. At the present time there are in existence ten sub-committees, each having accepted the duty of advising the Society through its Committee on Meetings not only of topics which should be treated in these respective fields, but also securing the proper treatment in papers and discussion. These committees comprise Textiles, Administration, Cement Manufacture, Iron and Steel, Industrial Building, Hoisting and Conveying, Railroads, Machine Shop Practice, Air Machinery, and Fire Protection. This plan should be continued until every branch of professional interest and public service shall be represented by a sub-committee that can give it whatever public expression it merits.

In the conduct of the two stated meetings of the Society, the Committee on Meetings is also enlisting the services of other sub-committees to take charge of the social and entertainment features, duties which, while only intermittent, involve very considerable labor and time. This year these features of the Annual Meeting are divided between the House Committee, which will take charge of all the details of the President's Reception, and the New York Local Committee, which will have the care of all other social sessions. In connection with the latter, your committee also reports the inauguration of the policy of payment for the social entertainment by those participating, all funds for this purpose being derived from the sale of tickets. It is hoped that this plan will work so successfully at this year's Annual Meeting, where it will have its first trial, that it can be likewise applied to the Spring Meeting, relieving local membership in various cities of the necessity of personal contributions for the entertainment of others.

Respectfully submitted,

Charles E. Lucke, *Chairman*
H. deB. Parsons
W. E. Hall
H. E. Longwell
H. L. Gantt

Committee on Meetings

REPORT OF THE COMMITTEE ON MEMBERSHIP

During the past year the Committee on Membership has held ten meetings for the transaction of its work, and in addition three joint meetings with the Committee on Increase of Membership. The number of applications acted upon is as follows:

Applications pending October 1, 1911	94
Applications received during fiscal year October 1, 1911 to September 30, 1912	771
	865

On these applications the following action was taken:

Recommended for membership	694
Withdrawn for various reasons	4
Deferred indefinitely	4
Deferred for further information	3
Applications held pending the receipt of endorsement of required number of references which did not arrive up to the date of closing	160
	865

It was also voted to reinstate eight members.

This committee, after extended investigation, recommends an additional grade of membership, with consequent modifications of the Constitution, to be voted upon at the Annual Meeting; and has also revised both membership applications and endorsement blanks with a view to omitting certain features which have been criticized, and simplifying the former to facilitate and reduce the cost of handling. The following comparison shows that some reduction in cost has already been secured:

In 1911, 368 candidates were placed on the Professional Service Sheets, at a cost to the Society of $5.81 each for the clerical work of the committee and for the ballot by the entire membership. In 1912, 694 candidates appeared on the Professional Service Sheets at a cost of $5.14 each.

Of the total expenditures ($3572.04) chargeable to this committee during the past year, $1925.15, or approximately 54 per cent of the total, was incurred for the printing, mailing and handling of the Professional Service Sheets and Ballots, which amount will be wholly saved under the new system of election and announcement now under consideration.

A further possible economy not yet fully investigated but perhaps worthy of consideration, may be effected by making a charge to new members for the certificates of membership now furnished free of charge. A preliminary search shows that the principal European scientific societies furnish their certificates without charge. Of the American Societies, the American Institute of Mining Engineers do not issue a certificate, the American Institute of Electrical Engineers charge $1 for a membership certificate, and the American Society of Civil Engineers furnish a paper certificate free of charge, but charge $2 for a parchment certificate.

Although the annual gain in members has been increased 88½ per cent over last year, a perusal of the last two ballots is most gratifying and convincing proof that the standard of membership has been fully maintained and will in no way be lowered by a dignified presentation to the mechanical engineering profession of this country of our claims to their membership and support, and the benefit to be derived by them from the constantly increasing lines of investigation and discussion in every branch of mechanical engineering presented at the meetings, and through The Journal of the Society.

Respectfully submitted,

G. J. Foran, *Chairman*
Hosea Webster
Theo. Stebbins
W. H. Boehm
H. C. Meyer, Jr.
Committee on Membership

REPORT OF COMMITTEE ON PUBLIC RELATIONS

The Committee on Public Relations having been requested by one of our Philadelphia members that the Society coöperate in matters of public interest, responded by notifying the local members in their individual capacity. While the Society, as such, may not always respond exactly as invited, it may in effect assist in relations of the engineer with the public, by corresponding with the members individually, inviting their attention to certain matters, and thus perform our duties in this regard.

Your committee would be pleased to receive communications and will do their best on all occasions to develop this activity of the Society's work.

Respectfully submitted,

James M. Dodge, *Chairman*
D. C. Jackson
John W. Lieb, Jr.
Fred J. Miller
Worcester R. Warner
Committee on Public Relations

REPORT OF THE PUBLICATION COMMITTEE

The adequate presentation and record of the growing work of the Society has imposed upon the Publication Committee an increasingly difficult task. It is impossible to present in a single volume of Transactions all of the papers presented before the Society at its two regular meetings and at the meetings of the several local groups and selections, as well as the record of Council and Committee meetings, administrative affairs, etc.

At the commencement of its term this committee decided with the approval of the Council, to confine the papers recorded in the Transactions to those presented at the regular spring and fall meetings, with a condensation of the discussion thereon, bringing within this category papers of exceptional merit presented at local or section meetings by referring them to the Committee on Meetings, with a recommendation that they be presented, at least by title, at one of the regular meetings; and further, that matter of administrative and only temporary interest would find sufficient publicity in The Journal and might be omitted from Transactions.

During the past year the number of papers presented at the regular meetings has been such that even with the discussion evoked condensed to the utmost, it is impossible to print them in a volume of less than 1500 or 1600 pages. The cost of such a volume would be $10,000, or $2000 in excess of the appropriation available for this purpose.

The committee requires an expression from the Council as to whether it may have the additional appropriation of $2000 necessary to print a volume which shall include all of the papers presented at the spring and fall meetings with the condensed discussion thereon.

There has been no general index of Transactions published previous to Volume 20. An appropriation of $1000 is now available for the purpose of bringing this index up to date, and steps will be taken to put the work immediately in progress.

The progress of The Journal has been most gratifying. It is the purpose of its editor, heartily concurred in by the Committee, to make it not only a means of communication between the membership and headquarters, and of distributing the papers presented before and the record of work accomplished by the Society itself, but an epitome of progress in the world of mechanical engineering indispensable to any active member of the profession. The Foreign Review, instituted during the past year is a step in this direction. It is hoped to extend this and kindred features until the contents of The Journal shall include at least directions to information upon everything of importance developed or presented in its field.

During the past year The Journal has earned $2000 over its expenses, taking care of the publication work incidental to (*a*) the increase of material presented at the various meetings, due to the formation of local committees and subcommittees, and (*b*) the increase in membership which includes the addition of over 1000 student members and affiliates since The Journal began to publish advertising, and (*c*) the Foreign Review which involves an expenditure of about $4000 per year. The present edition is 6500 with an average of 190 pages per month, as against an edition in 1907-1908 of 4000 with an average of 140 pages.

For the first time in the history of The Journal we have published some account of the papers and discussion at all meetings in which the Society participated, and synopses of the papers and discussion presented at all meetings held under the supervision of the Society whether locally or otherwise. The cost per 100 pages per 1000 copies has diminished from $135 at the time advertising was started to $120. The condensed catalogue plan of advertising has proved attractive and is more in keeping with the character of the Society's publication than the usual display advertising.

The Journal has proven its potency to serve the Society in many ways, with no draft upon the Society's funds, and is capable of being made a source of considerable revenue or of supporting itself in a wider field of activity and usefulness. By the publication of papers, committee reports, etc., in The Journal, the Society is now saved what was formerly a considerable item of expense for printing and distribution.

The Society must soon consider whether in view of the opportunities for publicity and record afforded by The Journal it desires to continue the publication of Transactions. At the present rate of increase it will soon be necessary to print two volumes per year. This will mean an annual expense of from $13,000 to $16,000. The two publications involve a duplication of expense and effort in many ways, and it is possible that a plan might be evolved whereby the binding of The Journal in whole or in part might serve all the purposes of Transactions with a considerable saving.

Respectfully submitted,

F. R. Low, *Chairman* G. I. Rockwood G. M. Basford C. I. Earll I. E. Moultrop	*Publication Committee*

REPORT OF RESEARCH COMMITTEE

Although the activities of the Research Committee have shown a decided increase during the year, it is still evident that the membership of the Society does not realize the existence of the committee and the power which it can exert for stimulating research in many directions, tending to advance the cause of mechanical engineering. The committee should be the medium by which the members of the Society bring to bear upon their problems the best assistance of the members of the profession throughout the country, and it is hoped that a greater realization of the possibilities of the committee in giving such assistance may be attained.

The committee has felt that the best way to bring these possibilities to the attention of the membership was to secure, first, statements from experts in the form of papers, abstracts, etc., of the present state of knowledge in various directions; and, second, to use these papers in the widest possible way to stimulate research to extend such knowledge. With this in view, the sub-committee on Steam Research secured a paper last year from Mr. George A. Orrok, on the Transmission of Heat through Metallic Tubes, an extremely able resumé of the present state of knowledge in that field; and has this year secured from the same author a continuation of this paper in the form of a digest of the technical literature on the transmission of heat. While this digest is not proper material for a paper to be presented before the Society, it is of the greatest value and the committee recommends that this paper be printed and thereby placed at the disposal of the members of the Society.

In addition the following papers on the following subjects are in preparation: The present status of the railroad rail question, with reference to the increased service demanded of rails and the improvements being made in rail construction, forming in fact a complete resumé of the entire question of breakage of railway rails and the steps being taken to relieve this condition in the future; deterioration or growth of cast iron in an atmosphere of superheated steam; the effect of pulsating flow on the accuracy of flow meters; steam heating of towns from central stations; amount of air and other gases in injection water.

There has been considerable discussion on plans for future activities, which will no doubt bear fruit in the succeeding year.

Research work is in progress on the coefficients of elasticity of ceramic materials, at the instigation of the committee on Mechanical Properties of Materials used in Electrical Engineering.

The following sub-committees have been appointed and have shown activity and devotion to the subjects assigned to them: Sub-Committee on Steam Research: L. S. Marks, C. J. Bacon, E. J. Berg, W. D. Ennis, J. F. M. Patitz, R. H. Rice; Sub-Committee on Research in Materials of Electrical Engineering: R. D. Mershon; Sub-Committee on Research in Safety Valves: P. G. Darling, E. F. Miller, H. D. Gordon, F. L. Pryor, F. M. Whyte.

Respectfully submitted,

Richard H. Rice, *Chairman*
R. D. Mershon
W. F. M. Goss
A. L. DeLeeuw
R. C. Carpenter
Research Committee

NECROLOGY

MAUNSEL WHITE

Maunsel White, for many years Engineer of Tests and Metallurgist of the Bethlehem Iron and Steel Works, died in New Orleans on October 22, 1912. He was born in Plaquemines parish, La., on March 15, 1856, of a prominent southern family. After a few years of study under private tutors and in Georgetown University, near Washington, D. C., he entered the school of technology at Worcester, Mass., and later the workshops of the Lehigh Valley Railroad at Weatherly, Pa. Here he gained a practical knowledge of mechanics which determined him to secure an engineering education. In 1875, accordingly, he became a student at Stevens Institute of Technology, from which he was graduated four years later as valedictorian of his class.

Immediately afterward, White was engaged by the Bethlehem Iron Company, under John Fritz, then largely manufacturing merchant steel and rails, now the Bethlehem Steel Company, and later became chief of the metallurgical department, taking high rank as an authority upon metals. He represented his company and was in charge of their exhibit at the World's Fair in Chicago in 1893, as well as at the Paris Exhibition of 1900, receiving personally at the latter a bronze medal of merit as a result of his company's exhibit; and he made frequent visits to Europe in the same capacity during his connection with the company, arranging among other things for the sale of great quantities of armor plate to the government of Russia, through personal interviews with the ministers of the Czar and at least one audience with the Czar himself.

Both the chemistry of iron and steel and the physical properties of these metals have been so thoroughly studied during the past 30 years and practice concerning them has become so crystallized, that the engineer of today finds it difficult to appreciate that facts which now seem elementary and commonplace were unknown in the early days of the industry. The life of the metallurgist in those days was intense and exciting. Problems were crowded on him upon which depended the very life and existence of his company, and in the solution of these problems, in wringing these secrets from nature, Maunsel White had a large though often inconspicuous part. The confidence inspired by his work is shown in the trust placed in him by the government officials detailed to supervise the manufacture of armor plate and ordnance at the time the Bethlehem Steel Company was so largely engaged in its construction. White's O.K. on a piece of steel was regarded as final.

Maunsel White will best be remembered through his part in the extended investigation which led to the discovery of the Taylor-White process of treating steel and the development of "high-speed" tools, which have revolutionized the machine shop practice of the world. In general he has been accorded the credit which belongs to him in the discovery of this process, but few have appreciated the detailed thoroughness of his work, that true scientific devotion which enabled him to plod along, month after month, developing methodically one fact after another, and the rare ability with which he was able, in the end, to coördinate and group these facts so as to indicate the conclusions and theories towards which they tended. High-speed tools have in many cases doubled and trebled the output of metal-cutting machines, and practically all makers of high-speed steel have today adopted for their best brand, steel of almost the exact chemical composition recommended by White as a result of his experiments.

While Maunsel White will ever be remembered for his metallurgical discoveries, his friends knew him as a man of rare versatility and catholicity of taste, with an intense love of literature and the arts and marked ability in these directions as well as in the realm of applied science. He was both a poet and a mathematician, a combination in the same individual so rare that in this respect he was a marvel to all his friends. Alive to beauty, he could at the same time get at the kernel of a problem with quickness, clearness and accuracy. His work at the Bethlehem Steel Company gained for him the friendship and admiration of John Fritz, the dean of the steel industry in America.

Mr. White was widely known both in America and abroad, and was a member of the Iron and Steel Institute of Great Britain and the American Institute of Mining Engineers.

PRELIMINARY REPORT OF THE COMMITTEE ON STANDARD SIZES OF CATALOGUES

The Committee has made an extensive investigation of the various sizes of catalogues in common use, and has had correspondence with many paper manufacturers, printers, advertising agents and makers of filing boxes and cabinets, as well as with manufacturers of machinery and supplies. The general conclusions it has reached may be summarized as follows:

a There is a universal opinion that catalogue sizes should be standardized. It is not necessary to give any reasons here for this opinion, as it is all on one side.

b The standard of 6 by 9 in. has for so many years met with such wide acceptance, probably two-thirds of all the catalogues that are now made being either that size, trimmed as closely to exact size as possible, or within ¼ in. of it in one or both dimensions, that it may be considered as too well established ever to be abolished.

c The 6 by 9 size is too small for many purposes, and it is necessary to have for some purposes a size which is about 8 by 10½ in. or 8½ by 11 in.

d Large manufacturing concerns have spent a great deal of money in having cuts and electrotypes made for these two large sizes, some using one size and some the other so that it is not to be expected that either size can be abolished. There is no serious objection to having both sizes remain as acceptable standards. Both of them are in use as standard sizes of letter paper, and both may be conveniently filed in the same filing case.

e The 9 by 12 in. size has been adopted by a few for very large catalogues. While it is generally acceptable for technical and trade journals, it is rather too large for a catalogue, unless it is to be a large

> cloth-bound book, to be placed on a shelf and not filed in a cabinet. It should, therefore, not be recommended as a standard for common use.

The following extracts from some of the letters received show the variety of opinions that exist as regards the sizes that should be adopted:

Matthews-Northrup Works, Buffalo, N. Y. Paper stock is made in sheet sizes 25 by 38 in. Since cuts have been made and tabular pages set for years to fit 6 by 9 in. it should prove very expensive for many firms to change. An opportunity is afforded by the 9 by 12 in. size to run the text descriptive matter or tabular panel of sizes and prices immediately under the illustrations on the same page. The 8½ by 11 size works out in good proportions. These three sizes seem to be the only ones that need be planned for.

Robert L. Stillman, New York. I suppose there are more catalogues made 6⅛ by 9¼ or 9⅜ by 12¼ than any other size, because they both cut from 25 by 38 in. paper which is of standard size, allowing ⅛ in. for trim.

The 33 by 46 in. size would make 8⅛ by 11¼. It might be reduced to 11 in. to get it within the standard letter size.

Ticonderoga Pulp & Paper Co., New York. It would be much to the advantage of every one if all catalogues could be standardized to a certain size. It would be a great assistance not only to the paper manufacturer but also to the dealer and printer, as in that case the dealer could carry in stock standard sizes and could supply paper promptly, and printers would not then have to wait for paper to be made especially.

Frank Presbrey Co., New York. Sent samples 6¾ by 9¾, and recommended it as the size used in all standard magazines. In reference to letter size, 8½ by 11, if a movement be inaugurated which will result in making up to this size all catalogues and monthly bulletins a great deal will be accomplished.

Collin Armstrong Advertising Co., New York. As far as our experience goes 5½ by 8 in. is the most convenient size. Anything near 8½ by 11 would in our opinion be too large to be handled easily, would be apt to go to pieces quickly and probably not be kept carefully by the recipient, as would be the case with a smaller book.

F. F. Coleman, Publicity Manager, Lidgerwood Mfg. Co., New York. All publications except catalogues intended for filing should be eliminated from consideration. The best size for catalogues is 9 by 12 in. upright. Personally, I would abolish the 6 by 9 size.

J. Horace McFarland, Harrisburg, Pa. In the nursery trade the 6 by 9 size does not permit such attractive illustrations in an economical manner as the growing vigor of selling campaigns required, and there is a great disposition to get away from anything of a standard size and to make the catalogue different.

Hammacher, Schlemmer & Co., New York. Are working on a new edition of their catalogue which will be 9 by 12 in. For a number of years they used 5½ by 7¾, but it is too small for a book of say 1200 pages. For pamphlets and smaller sizes will use 6 by 9 in.

Manning, Maxwell & Moore, New York. Books for general mechanical goods, both small and large should be 6 by 9 in. wherever it is practicable. In many cases a larger sheet is necessary to take the proper size cut to show important small details.

General Electric Company, Schenectady, N. Y. We use and recommend 8 by 10½ in. catalogues as standard. The width of page is valuable for displaying illustrations and tabular matter. Our standard letter paper is 8 by 10½ in. Or standard photographic plate is 8 by 10 in.

The Yale & Towne Mfg. Co., New York. C. L. Redfield, Adv. Mgr. Printed matter may well be any of the following sizes for reasons given: 6 by 3⅜ in. for folders, because this size just fits the small Government envelope which is still very largely in use among business people; 8 by 3½ in. for folders and small booklets, because it just fits the No. 9 Government 2 cent envelope and makes it very easy to use such printed matter to accompany correspondence; 6 by 9 in. for small catalogues, as we have been led to believe that this is more acceptable to the machinery and other trades; 8 by 10½ in. for sheets which accompany correspondence, as this is the size of our letter head; 9 by 12 in. for our more important catalogues, this being the accepted size with the paper makers and is carried in stock.

RECOMMENDED SIZES

	In.
Index Card, Standard	3x5
Index Card, Larger Sizes	4x6 and 5x8
Folders, Small 6x3⅜	Large, 8x3½
Catalogue, Standard Size	6x9
Bulletins and Large Catalogues, Standard	8x10½ and 8½x11
Bulletins and Large Catalogues Smaller than Standard	8x10½

In addition to the recommendation as to sizes the Committee would also recommend the following:

a For paper covered catalogues intended to be permanently filed, the edges, including the cover, should be trimmed to exact size. No fancy deckled edge or dark tinted paper should be used.

b Overlapping edges of the cover are permitted when the catalogue is bound in covers stiff enough to support its weight in boards or heavy card paper.

c The title should always be printed on the exposed back of the catalogue, even in very thin stitched books, for the purposes of identification.

d Every catalogue should have a date on its title page.

e A standard size index card 3 by 5 in. should be enclosed in each catalogue, with the title of the book and a brief statement of the character of its contents printed on it.

Catalogues, Extra Large, and Technical Press......9x12

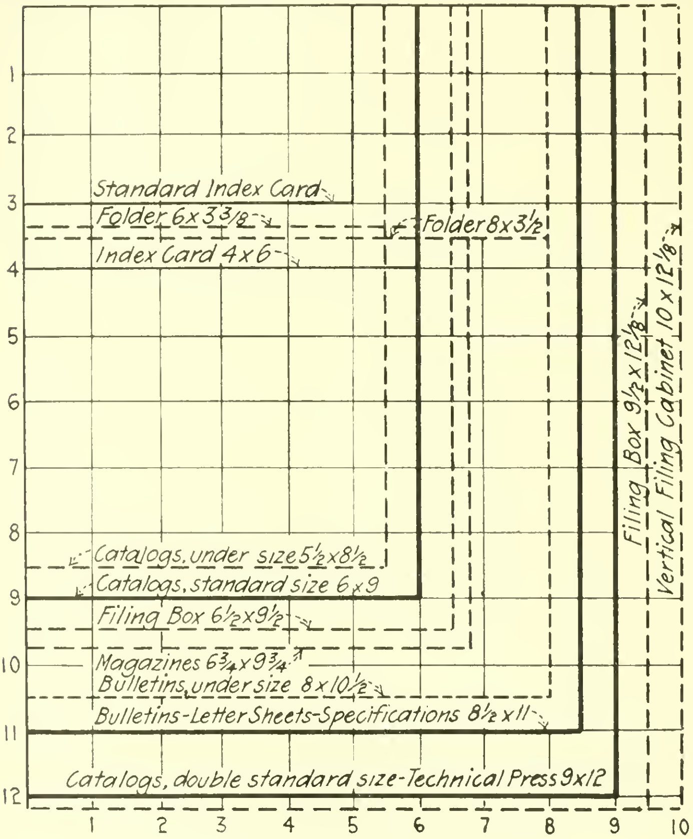
Standard Index Card
Folder 6x3⅜
Index Card 4x6
Folder 8x3½
Catalogs, under size 5½x8½
Catalogs, standard size 6x9
Filing Box 6½x9½
Magazines 6¾x9¾
Bulletins, under size 8x10½
Bulletins-Letter Sheets-Specifications 8½x11
Catalogs, double standard size-Technical Press 9x12
Filing Box 9½x12⅛
Vertical Filing Cabinet 10x12⅛
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FIG. 1 RELATION OF CATALOGUE SIZES

Transactions of Societies, Pamphlets, etc............6x9

PAPER BOXES, DRAWERS AND VERTICAL FILING CABINETS

(Inside Measurements)

For 6 x 9 in. catalogues, pamphlets and 9 x 12 papers folded once......................6½ x 9¾

Paper Boxes and Drawers for larger sizes....9½ x 12⅛

Vertical Files for large sizes................ 10 x 12⅛

Fig. 1 which gives a clear idea of the relation of these several sizes to each other is drawn to scale, showing them superposed.

FILING BOXES AND CABINETS

Jesse Jones Paper Box Co., Philadelphia, Pa.

A box that will hold a sheet 8½ by 11 in. is by far the most in demand.

Various makers of filing cabinets give the following sizes of drawers (inside measurements):

Correspondence Files 9⅞ x 11 9/16, 10 x 12, 10½ x 12⅛, 10 7/16 x 12.
Legal cap Files, 9⅞ x 14⅜, 9½ x 15, 10½ x 15¼.
Invoice Files, 7½ x 9½, 7¾ x 9¾, 8 x 10, 8¼ x 10 1/16.

The 9⅞ x 11 9/16 is too short to take a 9 x 12 in. paper. There seems to be no reason for making the height more than 10 in. as no letter paper is over 8½ in. wide. A size of 10 in. high by 12⅛ in. wide would seem to be the best for all large size catalogues. It would hold two 6 x 9 in. side by side, although the filing of 6 x 9 and 9 x 12 in. papers, together in the same drawer is objectionable, as the smaller sized papers have a tendency to lap over on each other and thus to occupy more space than they should. The extra inch in height gives ample room for division cards and tabs. The legal cap and the invoice files are not suitable for any of the standard sizes of catalogues. A size suitable for the 6 by 9 in. catalogue does not appear to be in the market. If one were made with drawers about 6½ by 9¾ in. it should have a ready sale. The drawer would be more easily handled than the 10 by 12⅛ in. size.

In submitting the above preliminary report the Committee desires to receive, from members of the Society and others interested in the subject, written discussions and criticisms of the recommendations. It is requested that if changes are suggested they be worded so that they may be inserted without modification in the revised report. When the preliminary report has been fully discussed and criticised the Committee will make a final revision for publication in the Transactions.

Respectfully submitted,

WM. KENT, *Chairman*
J. R. BIBBINS
M. L. COOKE
W. B. SNOW

Committee on Standardization of Catalogues

REPORT OF THE COMMITTEE ON STANDARD CROSS-SECTIONS AND SYMBOLS

We, the undersigned committee appointed by the President under the direction of the Council, to take up the matter of cross-sections and symbols, as suggested in a paper presented at the annual meeting of the Society in December 1911, beg leave to report that we have held meetings, have also carried on our work by correspondence, and now convey our unanimous findings as below:

We strongly believe that there should be recognition of a standard method of showing materials in cross-section. There are many advantages in encouraging the use of standard cross-sections and symbols. It is as easy to draw an adapted design to represent a specific material as to draw any other. It makes mechanical drawings easier to read and understand, and diminishes the danger of interpreting their meaning wrongly.

We do not believe it wise to complicate the matter by adopting too many standard cross-sectionings or symbols. We believe that it would be best to have standard cross-sections for the most commonly used materials, and that these should be of such a character as to permit of subdivision, if found desirable.

To this end, we propose the use of standard cross-sections to represent nineteen materials as shown on Fig. 1, viz., cast-iron, wrought-iron, cast-steel, wrought-steel, babbitt (or white metal), copper (brass or composition), aluminum, rubber (vulcanite or insulation), glass, wood, water, puddle, concrete, brick, rubble, ashlar, rock, earth and sand. To facilitate the drawing of cross-sections, the committee have used the same thickness for all lines made with the drawing pen.

On some drawings it may be desired to specify a material other than those mentioned above, or some particular kind of material which the above generic names would not clearly indicate. To cover this contingency, we recommend writing the name of such material on the section and cross-hatching the section, as shown on Fig. 1 under the title Other Materials.

We recommend that subdivisions of any of the materials shown

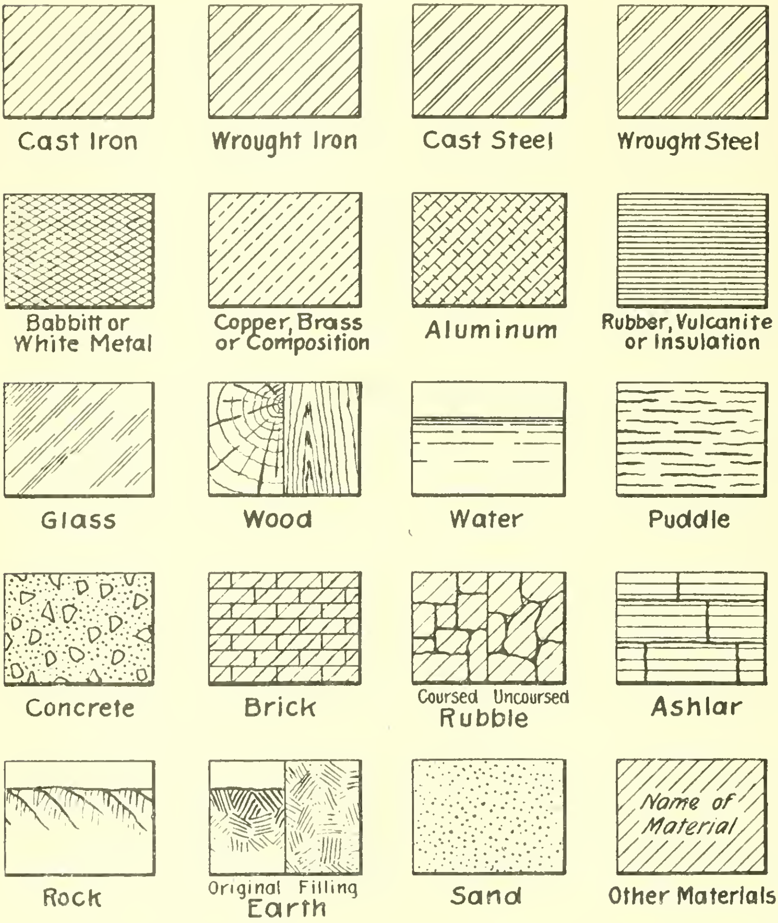

Fig. 1 Recommended Standard Cross-Sections

generically on Fig. 1, should be made by taking one of these standard cross-sections as a basis and making minor changes, but maintaining the general characteristics; or by writing on the standard section the name of the material. To illustrate, the committee have subdivided concrete into concrete-blocks, cyclopean-concrete and reinforced-concrete, as shown on Fig. 2, and also wrought-steel into nickel, chrome and vanadium steels.

We urge the Society to adopt standard cross-sections. Such standard cross-sections should be printed in a suitable form for hanging on walls of drafting rooms of engineers, architects, and educational institutions, so as to encourage their universal use.

Before closing, we acknowledge the valued assistance of Mr. D. C. Johnson in obtaining data for the committee and extend our thanks to the many manufacturing establishments, government departments and individuals who have sent us such standards as are in use by them.

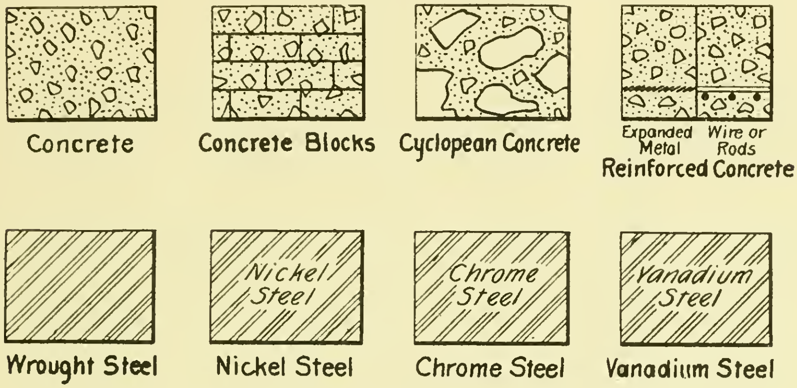

Fig. 2 Typical Subdivisions

Respectfully submitted,

H. de B. Parsons, *Chairman*
F. DeR. Furman
A. E. Norton
Bradley Stoughton
John W. Upp

Committee on Cross-Section and Symbols

EXPERIMENTS WITH NORTH DAKOTA LIGNITE IN A STEAM POWER PLANT AND A GAS PRODUCER

By Calvin H. Crouch

ABSTRACT OF PAPER

The North Dakota lignite fields cover approximately one-half of the State, or about 35,000 square miles. According to the State Geological Survey report, there were known to be in 1908 twenty-one workable seams of this coal in the southwestern part of the State, varying from 4 to 35 ft. in thickness and found in from 1000 to 1300 ft. of strata.

In this paper are reported some preliminary experiments showing the physical properties of the coal and the results when subjected to heat and the drying action of air; also two evaporative tests and 18 gas producer power plant tests, in which the coal was used as it came from the mine, except that it was broken so as to be charged easily into the gas producer.

The results of the evaporative tests, though not as satisfactory as could be desired, were interesting and instructive, indicating the possibilities of its use for such purposes and the improvements needed in furnace operation to burn it properly. The producer plant tests were highly satisfactory, showing that the coal could thus be utilized without special treatment and that the gas operated satisfactorily in the gas engine, giving a coal consumption in some of the longer tests of about 6½ lb. of coal as fired per kw-hr. at the switchboard for power available for outside purposes. If the power consumed by the gas scrubber was reduced to as small an amount as possible and the generator had been operated at more nearly its rated capacity, the results would have made a much better showing.

EXPERIMENTS WITH NORTH DAKOTA LIGNITE IN A STEAM POWER PLANT AND A GAS PRODUCER

By Calvin H. Crouch, Grand Forks, N. D.

Member of the Society

North Dakota has an area of approximately 70,000 square miles, of which about one-half is underlain with workable seams of lignite coal, varying from a few feet to 35 feet in thickness, which it is quite probable will some day exert a great influence upon its manufacturing and engineering industries. According to the State Geological Survey reports, the number of workable beds in southwestern North Dakota in 1908 were known to be 21. These are distributed through from 1000 to 1300 ft. of strata and range from 4 to 35 ft. in thickness. The aggregate thickness of the coal in these beds is 157½ ft.[1] Figs. 1 and 2 show the way in which the coal crops out[2].

2 It has been estimated by some that the North Dakota lignite deposits comprise one-sixth of the known coal deposits of the United States, but their extent and value are doubtless unappreciated, if indeed they are known, by the majority of the citizens of the State. Comparatively small amounts of the coal are mined annually, though the amount is increasing and will doubtless increase even more rapidly as the people learn to appreciate its value and to burn it in such a manner as to utilize more fully its heat units. It is regarded by many as an almost worthless fuel because it will not stand shipping in dry, warm weather without disintegrating and because it has been used in the ordinary furnace under boilers with poor results. It is used principally at points near the mines for power and domestic purposes and with

[1] N. D. State Geological Survey, 2d, 3d, and 5th biennial reports.

[2] N. D. State Geological Survey, 1908, Plates 13 and 15.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

the present cost at the mine and freight rates and the present method of burning, it does not seem able to compete with the cheaper grades of Eastern bituminous coal at points a few hundred miles from the mines. Its use, however, is increasing, as shown by a statement received from the state engineer's office that 271,928 tons were mined in 1904, 385,882 tons in 1910, and 466,600 tons in 1911.

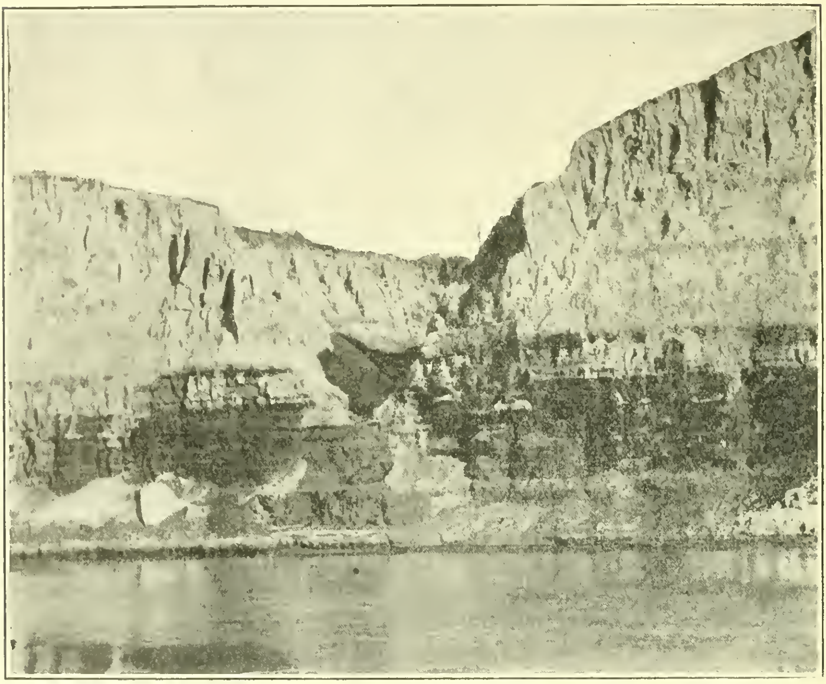

FIG. 1 COAL BED OF THE GREAT BEND GROUP EXPOSED ON LITTLE MISSOURI NEAR THE HARMON RANCH. TOTAL THICKNESS OF COAL, 16 FT.

PRELIMINARY EXPERIMENTS

3 Before presenting and discussing the results of the experiments with North Dakota lignite as a fuel in a steam and producer gas power plant, the author will present the results of some rather simple but interesting preliminary experiments with the same coals as were used in the power plant tests. They show the nature of the fuel and its behavior when exposed to heat and

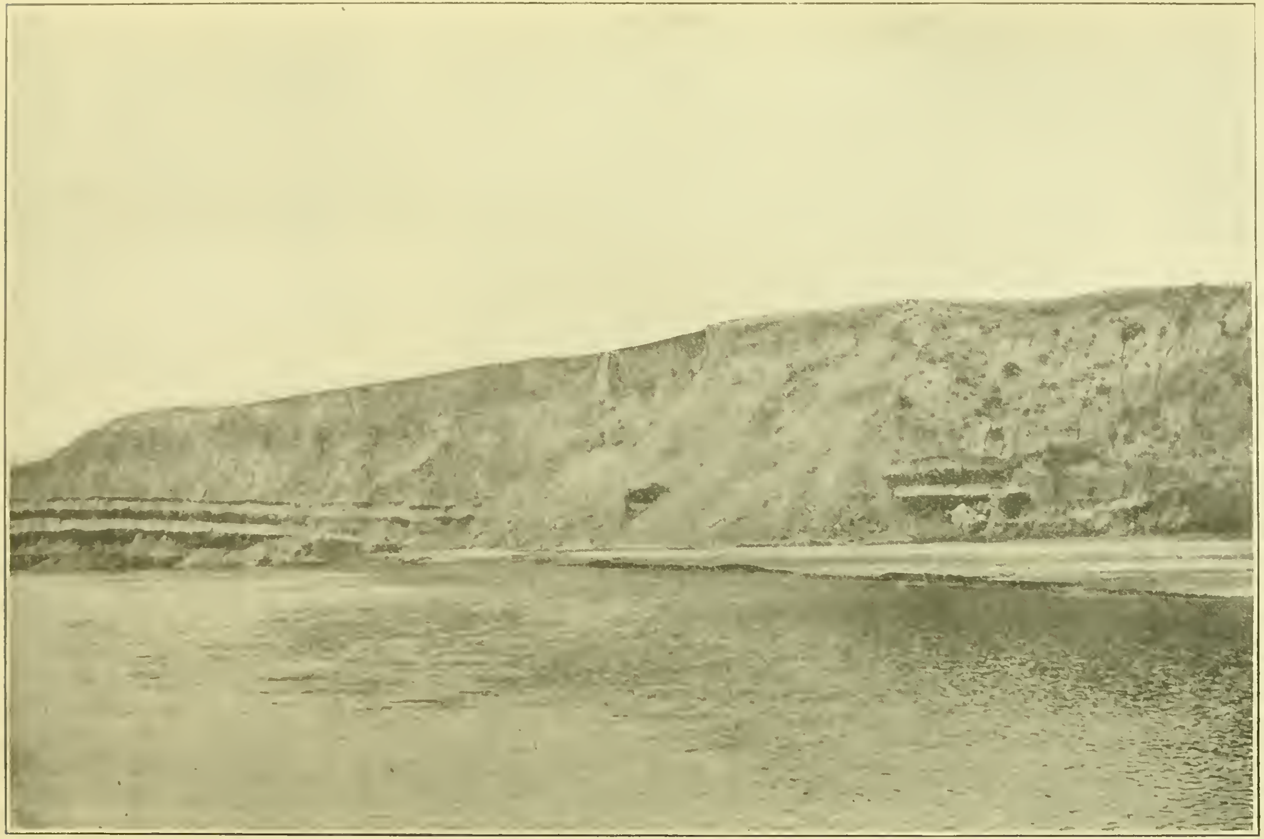

FIG. 2 COAL BEDS OF THE MEDORA GROUP EXPOSED ON LITTLE MISSOURI RIVER NEAR THE MOUTH OF ASH CREEK. TOTAL THICKNESS OF COAL, 16 FT. 5 IN.

to the drying action of the atmosphere and throw much light upon the results obtained in the power plant tests.

4 The coal used consisted of two car loads, one shipped in January 1911, from Williston, N. D., here designated as coal No. 1, and a second from Wilton, N. D., shipped in August 1911, referred to as coal No. 2.

5 Three samples of coal No. 1, weighing about 14 lb. each, were taken from different sections of the pile and weighed, submerged in water for 48 hours and again weighed. The gain in weight was but a fraction of 1 per cent, indicating that the coal was practically saturated with water when mined as it was shipped in cold weather and could not have given up or taken up much moisture in transit.

6 Coal No. 2 was shipped in August 1911, and was stored out of doors until needed for the tests. It was exposed to the elements during the summer and fall and naturally gave up some of its surface moisture, but the coal in the interior of the pile was quite wet in the winter when the tests were made. A block of this coal weighing 12 lb. 10 oz., which was taken from a point 2 ft. beneath the surface of the pile, submerged in water for 48 hours and again weighed, gained 14 oz., or 6.9 per cent in weight. The author was not present when coal No. 2 was received, hence, samples were not taken at that time.

7 To show the readiness with which the lignite gives up its moisture, four samples weighing respectively 30, 38½, 74 and 83 lb. were taken from coal No. 2 and placed in a room on March 16, 1912, and weighed from time to time. During the first week, the temperature was about 60 deg. fahr. and during the remainder of the time about 70 deg. On March 18, the coal had given up 2.43 per cent of its weight, on March 20, 3.76 per cent, on March 21, 6.09 per cent, on March 27, 9.2 per cent, April 1, 11.31 per cent and on April 16, 12.63 per cent. Figs. 3, 4, 5 and 6 were taken of these four samples on March 18, 20, 27 and April 16 and show the way in which the coal disintegrated as it dried. Fig. 7 shows the results after these four samples had been dropped 8 ft. upon a concrete floor, and indicates that when the coal is dry it is very friable.

8 To show how the coal disintegrates when stored for some time under cover, on April 6, 1912, a sample of 100 lb. was taken from a bin inside the building which had been filled with coal

No. 2 since August 1911. The bin contained about 4 tons. A small proportion of the coal (perhaps one-sixth) was in large chunks ready to fall into pieces with a light blow. The coal had been pulverized somewhat by firemen in shoveling it from the bin. However, neglecting the large chunks, the sample was a fair average of the coal. This sample was screened successively on sieves with ⅛ in., ¼ in. and ½ in. mesh; that which remained on the finer sieve was screened on the next coarser. Of the coal 26½ per cent passed through the screen with the ⅛ in. mesh, 24½ per cent passed through the screen with the ¼ in. mesh, 18 per cent passed through the screen with the ½ in. mesh and 31 per cent remained on the last. All would have passed through a 2½ in. ring.

9 Numerous proximate analyses were made, the results of which showed the green coal to contain approximately 35 per cent of moisture, and the dry coal to contain approximately 43 per cent of volatile matter, 50 per cent of carbon and from 7 to 9 per cent of ash. Difficulty was experienced in determining the percentage of moisture. It was finally decided to call that moisture which was driven off at a temperature below 240 deg. fahr. It was thought that some of the volatile matter was driven off at temperatures slightly above that. It was readily seen that gases were driven off at comparatively low temperatures as shown by the analytical balance and by the odors given off.

10 North Dakota lignite is classed as one of the youngest coals. In appearance it is black when freshly mined, but as it dries, it becomes lighter in color and is often a dark brown. Before it has become sufficiently dry to disintegrate, it somewhat resembles charcoal in that it frequently shows very clearly the grain of the wood and knots of the trees from which the coal has been formed; but upon drying it cracks both with and at right angles to the grain as seen in Figs. 3-7. The line of cleavage in breaking is along the grain. Occasionally a piece will be found which has lost its grain and at fracture resembles anthracite in appearance, having a luster, but this disappears as the moisture is driven off.

11 From these preliminary experiments, it will be seen that this fuel may contain (*a*) large quantities of moisture, the two coals analyzed containing over 35 per cent of moisture, which is readily given up, causing the coal to crack and disintegrate; (*b*)

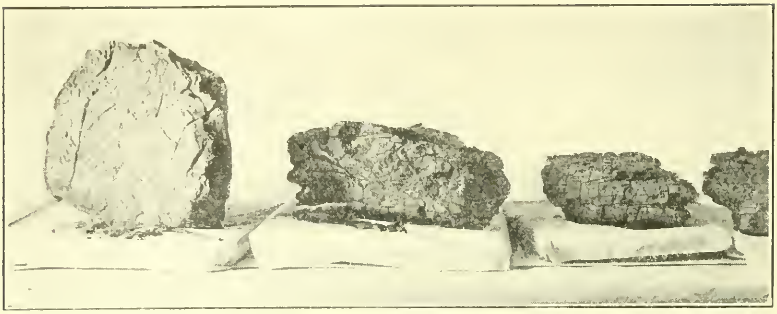

Fig. 3 Coal after being 2 Days inside of Building

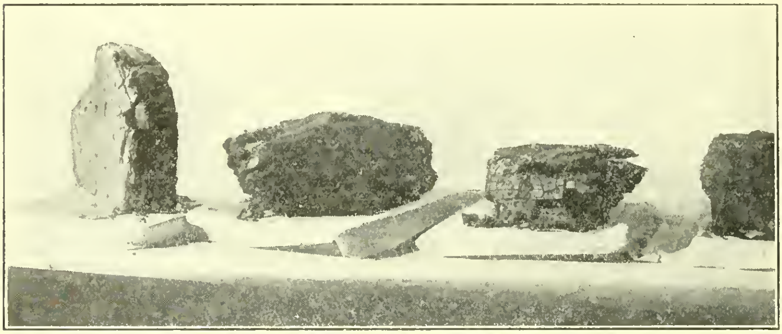

Fig. 4 Coal after being 4 Days inside of Building

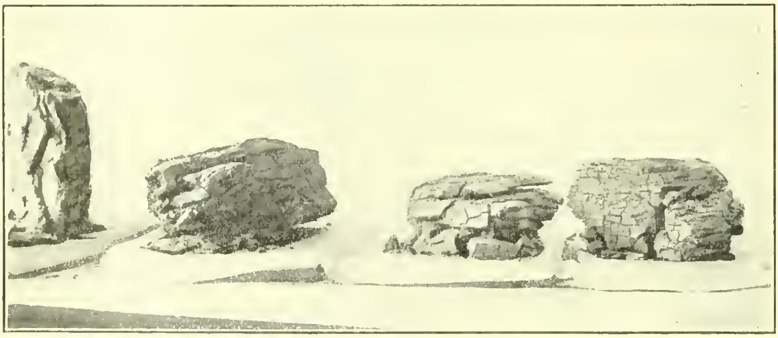

Fig. 5 Coal after being 11 Days inside of Building

large proportions of volatile matter, some of which is driven off at comparatively low temperature; and (*c*) a small percentage of ash. These characteristics are very important and must be taken into consideration when designing furnaces and gas producers to utilize it. In the furnace it disintegrates and if burned on grates suitable for a coking coal, it will, when heated to red. run through the grate like so much coarse sand, especially if it be disturbed with a slice bar. Thus it will be seen that it should be burned on a grate with small openings and with a strong draft, for when it disintegrates it settles or packs down, restricting the air passages. One would also expect, since it contains a large proportion of volatile matter, that fire-brick arches or dutch ovens would be desirable in burning it under boilers.

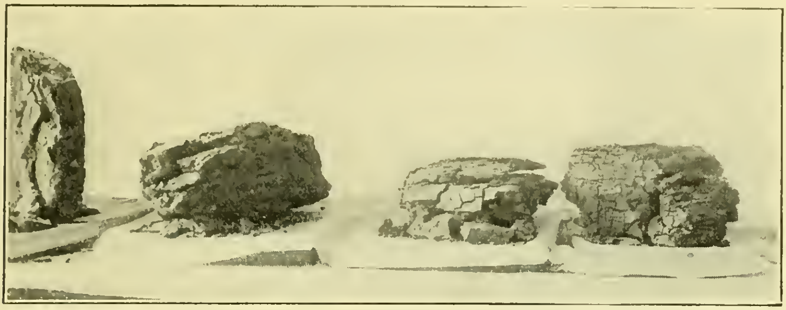

Fig. 6 Coal after being 31 Days inside of Building

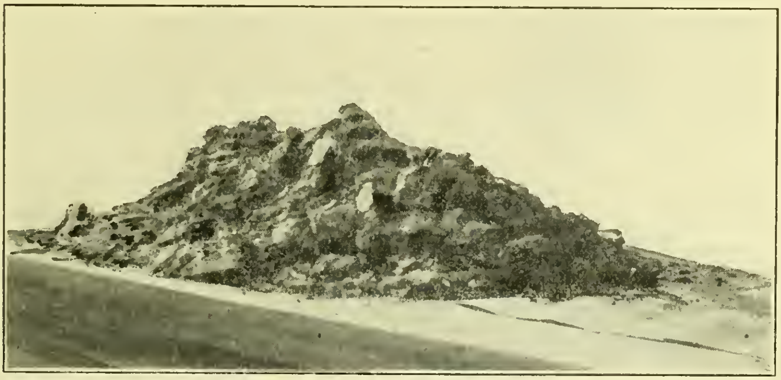

Fig. 7 Samples of Coal after falling 8 Ft. upon a Concrete Floor

12 An appreciation of these important properties and characteristics shown by the preliminary experiments will help in understanding the difficulties experienced in conducting the evaporative and gas producer tests and interpreting the results.

EVAPORATIVE TESTS

13 Evaporative tests, using North Dakota lignite under the boilers, were made in the mechanical laboratory of the College of Mechanical and Electrical Engineering at the State University of North Dakota. This plant consisted of two 70-h.p. horizontal fire tubular boilers, one of which was equipped with a Detroit automatic stoker and a dutch oven 7 ft. long projecting out in front of the boiler, while the other was provided with an ordinary furnace and rocking grates which were suitable for burning eastern bituminous coals. After several preliminary experiments, this last-mentioned furnace was modified by placing fire-brick arches or tiles over three-fourths of the grate and by substituting perforated plates for the rocking grates, the perforations being ½ in. in diameter and 35 per cent of the area of the grate.

14 The plant was provided with natural draft which was adequate when burning Eastern bituminous coal to drive the boilers much beyond their rated capacities, but which proved inadequate when burning lignite. Several tests were conducted, varying from four to ten hours in duration. While they were unsatisfactory in that the boilers could not be driven at their rated capacities, they indicated conclusively that a much stronger draft would be necessary to keep the furnaces at a white heat and to force the boilers to their rated capacities. A small fan was available for producing a forced draft. It was not large enough to supply the necessary pressure but served its purpose for, with it, the furnace could be kept at a white heat, which was impossible without it. It was not thought advisable, however, to drive the fan at the necessary high speed for any considerable time and the results obtained with natural draft had to suffice. The author, wishing to confirm the results secured in 1911 by the students, conducted three tests in the spring of 1912, with about the same experience and results. Funds were not available at the time for the installation of an induced draft so that elaborate tests were not undertaken, the experiments being valuable chiefly because of the experience and knowledge gained. It was clearly

demonstrated that with a dutch oven and a mechanical stoker with proper draft conditions, the coal can be economically utilized in a steam power plant.

15 Table 1 gives the more important economic results obtained from these preliminary evaporative tests. For comparison the average of the results of 15 tests made by the U. S. Bureau of Mines, with this coal in a specially constructed furnace has also been included.[1]

TABLE 1 EVAPORATIVE TESTS

Furnace	Grate Area, Sq. Ft.	Duration of Tests, Hours	Per Cent Moisture in Coal	Dry Coal burned per Sq. Ft. of Grate Surface per Hour	Equivalent Evaporation from and at 212 Deg. per Lb. of Coal as fired	Equivalent Evaporation from and at 212 Deg. per Lb. Dry Coal	H.P. in Per Cent of Builders' Rating
Detroit stoker and dutch oven.....	20.25	10	35.77	12.20	4.22	6.58	67.4
Perforated plate grates with arches over 3/4 of the grate....	16	12	35.77	14.58	3.638	5.66	54.7
Special furnace used by Bureau of Mines.......	....	....	43.27	25.18	3.49	6.09	73.9 to 103.3

16 It will be noted that although the boilers were forced as hard as possible in the tests conducted by the author, they developed only from 54.7 to 67 per cent of the builders' rated capacity. In burning the coal on the chain grates under water tube boilers in the university power plant which had natural draft, a similar experience resulted. The experiment was begun with the furnace at a white heat, but as the lignite was fed into the furnace the fire-brick arches and tile on the water tubes cooled to a much lower temperature and the boiler could not be driven at anything like its normal rating.

17 An induced draft system was recently installed in the university power plant, and an induced draft apparatus has been ordered for the experimental plant in the mechanical laboratory

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[1] Bulletin No. 2, Bureau of Mines, pp. 18, 19.

which will be capable of giving almost any amount of draft that may be desired in connection with a series of tests which will doubtless be conducted in the near future.

GAS PRODUCER TESTS

18 Experiments with North Dakota lignite in the gas producer were conducted in the same laboratory, the object being to determine the feasibility of using this fuel in a producer gas power plant.

APPARATUS

19 The experimental plant consisted of a 50-h.p. Smith suction down-draft gas producer and a 55-h.p. Alberger tandem producer gas engine belted to a dynamo. The scrubber was driven by an independent motor. In the tests of 1911, the jacket and scrubbing water was supplied from the city mains but for the 1912 tests a motor-driven triplex pump was installed so that the cost in power of supplying this water could be determined. In the 1911 tests, the dynamo used to load the engine was a 25-kw. direct-current machine, but for the subsequent tests a 45-kw. direct-current three-wire generator with compensating coils was installed so as to be able to load the engine to its maximum capacity. Fig. 8 shows a sectional view of the gas producer and Fig. 9 a sectional view of the engine.

METHOD OF LOADING THE ENGINE

20 The engine was loaded by means of a dynamo which supplied current to the university light and power circuit and also to a water rheostat which could be used when there was not sufficient demand otherwise for power to utilize all that was produced. The dynamo was so connected that current could be taken from the main line to drive the generator as a motor in starting up the engine. This made a simple and very satisfactory way of starting the engine which, when started, would speed up, increasing the generated electromotive force, and cause the motor to become a generator supplying current to the line, the load carried by the dynamo being governed by the voltage which was controlled by a hand-regulated field rheostat.

21 The generator used during the tests Nos. 1-8 inclusive was an old 25-kw. direct-current generator formerly used in the university lighting plant; in tests Nos. 9-18 inclusive a new Fort

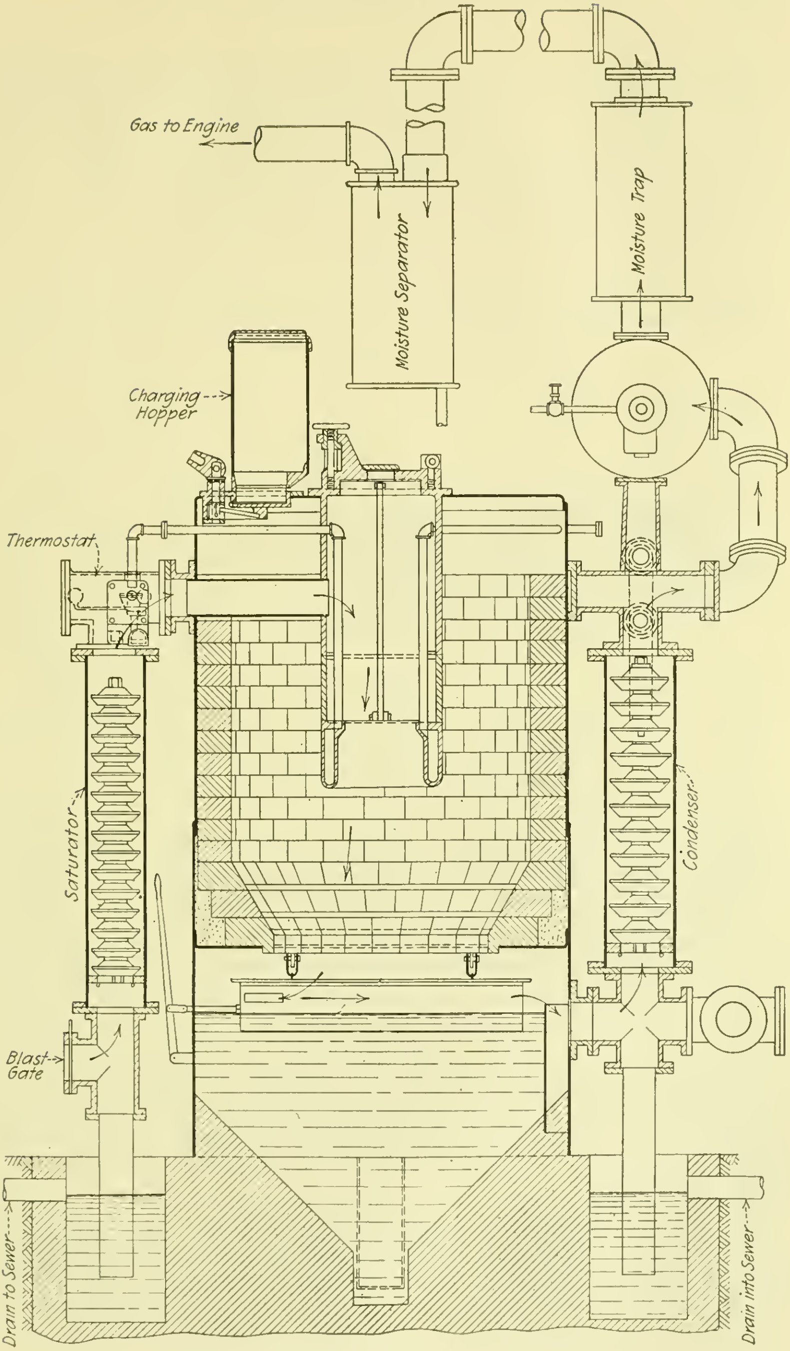

FIG. 8 SECTIONAL VIEW OF GAS PRODUCER

Wayne machine. This machine started up nicely but before test No. 12 had been finished, the commutator had become warped or out of true, causing much sparking at the brushes. This was remedied, however, before test No. 13 was run by turning down the commutator, after which it gave no trouble.

METHOD OF DETERMINING THE POWER GENERATED, AND COAL CONSUMED

22 The power delivered to the switchboard by the generator was measured, and the coal consumed per kw-hr. at the switchboard determined. It was the original intention to determine the efficiency of the generator so as to get the total power developed by the engine, but time did not permit before the armature of the generator was needed to replace a burned-out armature in a motor in the university power plant, so that the windage, friction and internal losses of the generator used in 1911 could not be obtained. However, since the machine was loaded to nearly its rated capacity most of the time, it was assumed in determining the brake horsepower of the engine that the generator had an efficiency of 90 per cent and that the belt transmission gave an efficiency of 98 per cent, both of which are probably higher than their true value.

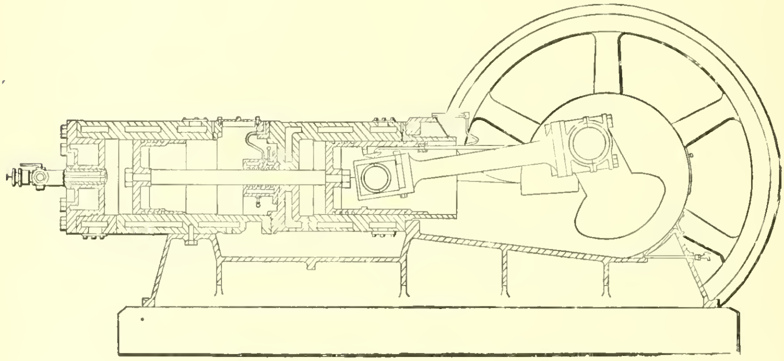

Fig. 9 Longitudinal Section of Gas Engine

23 As no means was at hand for measuring or determining the amount of water used by the engine jackets or the gas scrubber in the tests of 1911, accurate records of the power used by the

scrubber were not taken. To determine the net power available for outside purposes, it would be necessary to charge against such a plant all of the power consumed by the auxiliary apparatus. Fortunately funds were available in the fall of 1911 so that a motor-driven triplex pump was installed to supply the jacket and scrub water, also a more efficient motor was installed for driving the scrubber. In determining the net power available for outside purposes in the tests of 1911, it was assumed that the power consumed by the auxiliaries was the same as during the tests of 1912 when apparatus was available.

24 In determining the power developed by the engine in the tests of 1912, it was assumed that the belt transmission gave an efficiency of 98 per cent. The losses in the generator were determined by experiments in which the windage, friction, iron and copper losses were carefully determined and curves plotted showing the efficiency of the generator at various loads. The low efficiency of the generator may doubtless be explained by the fact that the machine was developing but little more than 50 per cent of its rated capacity.

25 In determining the coal used during the test, care was taken that the producer was full when the test began and when it ended. The coal charged in the meantime was considered as the coal used during the test. Before starting up the blower for the next test, the producer was poked to make sure there were no large voids and filled as before. The coal charged at this time was called the " standover " losses. A similar method was followed after fire had been " blown up " and the amount needed to fill the producer just before starting the test was called the " starting-up " losses.

COAL AND GAS ANALYSES

26 At the beginning of the first series of tests, Nos. 1-8, Table 2, the apparatus for analyzing the gases was broken so that but one analysis of the gas of these eight tests was made. Proximate analyses of the coal used in each test were made to determine the moisture. In the second and third sets of experiments, or those conducted in 1912, a complete analysis of the gases was made. The incomplete determinations of CO_2, O and CO were made with the Orsat flue gas apparatus. It is interesting to note that the composition of the two coals was almost identically the same.

TABLE 2 GAS PRODUCER PLANT POWER TESTS

No. of Test	Duration, Hr.	Coal Used	Moisture in Coal, Per Cent	Dry Coal Used	Moisture in Coal, Per Cent	Dry Coal Used	Coal as Fired per Hour	Dry Coal Used per Hour	Kw. Delivered at Switchboard	Kw. Consumed by Scrubber and Pump	Kw. at Switchboard Available for Outside Purposes	Power Consumed by Generator, H.P.	Brake H.P. of Engine Assuming an Efficiency for the Belt Transmission of 98 Per Cent	Coal as Fired per Kw-Hr. at Switchboard
1	8	632	32.9	424.1	32.9	424.1	79.0	53.0	14.18	3.71	10.47	.....	16.07	5.57
2	8	1017	33.1	680.4	33.1	680.4	127.1	85.0	21.21	3.71	17.50	.....	24.04	5.99
3	7¼	875	35.9	560.9	35.9	560.9	120.6	77.2	20.86	3.71	17.15	.....	23.65	5.77
4	27½	2575	16.0	2163.0	16.0	2163.0	93.6	78.6	22.64	3.71	18.93	.....	25.66	4.13
5	4	425	35.8	272.9	35.8	272.9	106.2	68.2	21.24	3.71	17.53	.....	24.08	5.00
6	8	850	38.0	527.0	38.0	527.0	106.2	65.8	21.27	3.71	17.56	.....	24.11	4.99
7	16	1700	36.5	1079.5	36.5	1079.5	106.2	67.4	22.67	3.71	18.96	.....	25.70	4.68
8	6¼	500	37.2	314.0	37.2	314.0	76.9	48.3	18.5	3.71	14.79	.....	20.97	4.10
9	24	2960	20.4	2356.0	20.4	2356.0	122.3	98.1	18.3	3.71	14.59	28.71	29.21	6.68
10	9½	1255	20.4	999.0	20.4	999.0	131.5	105.1	19.36	3.71	15.65	30.16	30.78	6.78
11	11½	1365	20.4	1086.0	20.4	1086.0	118.7	94.1	19.87	3.71	16.16	30.88	31.49	5.98
12	12	1125	20.4	895.0	20.4	895.0	93.7	74.5	19.12	3.71	15.41	29.82	30.43	4.90
13	9¾	1205	35.77	774.0	35.77	774.0	123.6	79.38	22.78	3.71	19.07	34.94	35.66	5.42
14	5 1/3	1072	33.0	718.0	33.0	718.0	201.1	134.7	21.31	3.71	17.60	32.91	33.58	9.43
15	9	1478	23.35	1132.0	23.35	1132.0	164.2	125.7	22.99	3.71	19.28	35.03	35.74	7.14
16	11	1351	23.35	1035.0	23.35	1035.0	122.8	94.0	22.88	3.71	19.17	34.92	35.64	5.36
17	7 2/3	820	23.35	628.0	23.35	628.0	107.0	82.0	24.54	3.71	20.83	37.37	38.13	4.40
18	24	3013	23.35	2309.0	23.35	2309.0	125.0	96.2	23.18	3.71	19.47	35.45	36.17	5.39

TABLE 2—Continued

No. of Test	Coal as Fired per Kw-Hr. Available for Outside Purposes	Coal as Fired per Brake Hp-Hr.	Dry Coal per B.H.P-Hr.	CO_2, Per Cent	O, Per Cent	CO, Per Cent	H, Per Cent	CH_4, Per Cent	N, Per Cent	Number of Samples Analyzed	B.t.u. per Cu. Ft.	Suction in Inches of Water at			
												Engine Throttle	Scrubber	In Pit below Grate	Air Inlet
1	7.56	4.92	3.29	....	....	....	....	....	....	....	....	....	....	....	....
2	7.26	5.28	3.53	....	....	....	....	....	....	....	....	....	....	....	....
3	7.02	5.10	3.26	....	....	....	....	....	....	....	....	....	....	....	....
4	4.94	3.65	3.06	5.10	4.65	17.58	9.31	2.3	61.06	....	111.4	....	....	....	....
5	6.05	4.41	2.83	....	....	....	....	....	....	....	....	....	....	....	....
6	6.05	4.40	2.72	....	....	....	....	....	....	....	....	....	....	....	....
7	5.61	4.13	2.62	....	....	....	....	....	....	....	....	....	....	....	....
8	5.20	3.66	2.30	....	....	....	....	....	....	....	....	....	....	....	....
9	8.38	4.22	3.35	11.9	2.4	11.6	....	....	....	2	....	15.8	13.6	4.9	0.25
10	8.40	4.27	3.41	9.2	2.5	16.7	....	....	....	2	....	20.7	15.2	7.3	0.7
11	7.34	3.77	3.00	10.8	2.3	14.0	....	....	....	3	....	21.9	17.4	7.6	0.1
12	6.09	3.08	2.44	10.0	2.6	12.5	....	....	....	4	....	21.1	16.8	10.7	0.18
13	6.00	3.46	2.22	9.8	1.0	10.8	....	....	....	3	....	23.0	20.0	6.0	0.2
14	11.42	5.99	4.01	....	....	....	....	....	....	....	....	23.6	18.0	4.61	0.7
15	8.52	4.59	3.51	....	....	....	....	....	....	....	....	23.4	19.1	4.4	0.5
16	6.40	3.44	2.63	9.13	1.64	14.11	16.53	0.88	57.7	12	109.85	18.2	16.0	7.5	1.27
17	5.13	2.8	2.15	6.08	3.8	10.62	18.97	4.2	56.25	4	152.7	26.9	20.8	4.7	0.07
18	6.45	3.47	2.65	8.59	3.7	11.7	8.04	4.77	63.18	9	115.42	28.8	21.7	7.5	0.33

Tests Nos. 1–8 were made between February 2 and May 24, 1911; tests Nos. 9–18 between February 6 and May 10, 1912. In all of these a Smith suction down-draft 50-h.p. gas producer and an Alberger tandem 55-h.p. gas engine were used; while in Nos. 1–8 a Westinghouse 25-kw. and in Nos. 9–18 a Fort Wayne 45-kw. generator was employed. The lignite coal in Nos. 1–8 was from the J. Bruegger mine, Williston, N.D., and in Nos. 9–18 from the Washburn Mine, Wilton, N. D. An analysis of samples taken from piles out of doors showed for coal No. 1, volatile matter, 44.2 %; fired carbon, 50.1 %; ash, 5.7 %; moisture, 35.9 %; for coal No. 2, volatile matter, 43.58 %; fixed carbon, 50.33 %; ash, 6.09 %; moisture, 35.77 %.

DESCRIPTION OF TESTS

27 Test Nos. 1-8 inclusive were conducted at intervals of from one to two weeks apart. After a test had been run, the producer was filled and enough coal brought into the building during the following week to run the next test. This enabled much of the coal to thaw out, though frequently coal was charged into the producer in the frozen state. The coal used in test No. 4, however, had been in the boiler room for some weeks which caused it to be much drier than that used in the other seven tests, but the author is inclined to believe that the 16 per cent moisture recorded is much too small. It is possibly due to poor sampling or to the loss of moisture between date of sampling and date of analyzing.

28 These tests were conducted with the idea of getting the maximum power possible out of the engine, but the capacity of the generator limited this, while tests Nos. 8-12 were planned to determine the efficiency of a plant, such as an electric light plant where the load would be variable and the plant operating from but 6 to 12 hours per day, standing idle the remainder of the 24 hours. For this reason, little effort was made to keep the load uniform. Ammeter and voltmeter readings were taken at half-hour intervals, but the power consumed during these four tests was measured by means of a carefully calibrated watt-hour meter. The power developed during tests Nos. 1-8 was determined from the ammeter and voltmeter readings which were taken every 15 minutes; during tests Nos. 13-18 inclusive, the power was likewise determined but the readings were taken at intervals of 20 minutes each, the loads in tests Nos. 1-8 and in 13-18 being kept as nearly uniform and as large as seemed feasible.

TROUBLES EXPERIENCED

29 It was observed that at the engine throttle, in the scrubber suction and in the pit beneath the grate of the producer, the suction increased as the tests progressed, apparently due to fine ash and clinkers closing up the air passages. This would restrict the gas at the engine causing its speed to reduce, which in turn would lower the voltage and cause the load to be thrown off automatically.

30 On one occasion the suction became so great that air

was pulled down through the ash conveyor into the bottom of the producer, with a resulting explosion that shook the fire up violently, blowing much fine ash out through the top of the central tube, which happened to be open. The engine automatically threw off its load, the dynamo operating as a motor driving the engine until the spent gases had been exhausted, when the engine immediately speeded up and took on its normal load. The suction at the different parts of the plant was greatly reduced after this explosion so that it operated satisfactorily during the remainder of the test.

31 Another source of trouble was that the strong suction would pull ash and fine particles of coal over into the bottom of condensing chamber which gradually became clogged and finally shut off the supply of gas with the natural results.

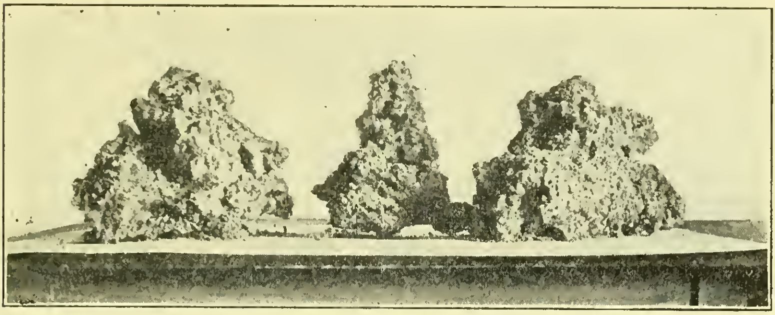

Fig. 10 Clinkers taken from Gas Producer

32 It is possible that with more experience the producer might be operated to fuse practically all of the ash into large clinkers, which could be removed through the top of the central tube of the producer. A large part of the ash was thus fused but evidently enough escaped to cause trouble. Fig. 10 shows some of the clinkers removed.

33 The disintegrating action of the coal when subjected to a high temperature caused the coal to settle down and the bed to become dense with restricted air passages, which also tended to increase the suction. Violent shaking of the grate seemed to help reduce this high suction and occasionally the suction would drop to but a few inches when large clinkers were removed.

34 It was observed in conducting the preliminary tests of

1912 when running tests on successive days using fuel with a large percentage of moisture, that the engine would satisfactorily carry its load during the first five or six hours and then the fire would become chilled and the gas so poor that the engine would scarcely run with no load. It was also observed when this happened that the fire in the producer had burned unevenly, being hotter on one side than on the other, doubtless allowing air to pass down through the bed without burning the coal. This action can perhaps be explained by the fact that the coal which had stood in the producer over night had become sufficiently dry to burn satisfactorily and that the trouble began when the fresh coal charged during the test reached the fire zone, chilling the fire because of its large amount of moisture and causing it practically to die out in certain sections and to become black at the lower end of the central tube where the fire was usually bright when operating under normal conditions.

35 The trouble is one that can be easily overcome either by drying the coal before shipping from the mine or by storing it for a few weeks in a dry, warm place where it will readily give up a large part of its moisture.

REMARKS

36 The starting and standover losses of the producer plant were surprisingly small. From a number of records made, it was estimated that the starting-up losses did not exceed 50 lb. even when the producer had been standing idle for one or two weeks and required 35 to 45 minutes to get good gas. The starting-up losses when the producer was operated on successive days were much less as good gas would be delivered at the engine in from 10 to 20 minutes from the time the blower was started. The engine would often carry full load within 15 or 20 minutes after starting the blower.

37 The standover losses for fourteen different observations covering 867 hours amounted to 3043.5 lb. or an average of 3.3 lb. per hour. The periods of standover varied from 12 to 300 hours, the standover losses for the short periods being more per hour than for the long periods. Taking 777 hours of the longer standover periods, the average per hour of the standover losses was but 1.92 lb. per hour. Comparing the starting-up and standover losses of the producer gas plant with those of a steam

plant, the author believes the former has a decided advantage over the latter.

38 It should also be borne in mind that a commercial plant would doubtless be much larger than this experimental plant and would give correspondingly better results. Furthermore, it is quite possible that the power consumed in scrubbing the gas was much larger than was necessary, as it was feared trouble would arise from tar getting into the engine.

39 Doubtless a larger amount of scrub water was used than was necessary, thus greatly increasing the amount of power consumed in scrubbing the gas. The amount of power consumed by the scrubber could be doubled or trebled simply by increasing the quantity of scrub water used. No trouble was experienced, however, from tar getting into the engine cylinders. The cylinder head of the engine was removed for inspection from time to time. A small amount of gummy substance adhered to the counterbore of the cylinder but the cylinder barrel was well lubricated. It was thought that the substance adhering to the counterbore was largely cylinder oil and carbon partially burned, but it may have been partially burned tar. In either event, there was not enough to cause the piston rings to stick or leak or to interfere with the action of the admission and exhaust valves.

40 A lengthy discussion comparing the economic results of the coal consumed per brake horsepower or per kw-hr. at the switchboard with the amounts required by steam plants of similar size is not necessary. It should be apparent to all that the cost of power in terms of pounds of lignite per b.h.p. or per kw-hr. at switchboard is much less than it would have been in a steam power plant of like size.

41 None of those interested in these tests was familiar with the operation of gas producers so that all had much to learn about the care and manipulation of the plant. Doubtless much better results would have been secured by those experienced in the handling of this type of plant. The author, however, feels gratified with the results for it should be expected that with more experience in handling the producer a much better economy would be obtained.

ACKNOWLEDGEMENT

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AXIOMS CONCERNING MANUFACTURING COSTS

By Henry R. Towne

BRIEF

In a paper entitled The Enginner as an Economist, contained in Volume 7 of the Transactions, 1886, by the present writer, is the following statement:

"To insure the best results, the organization of productive labor must be directed and controlled by persons having not only good executive ability, and possessing the practical familiarity of a mechanic or engineer with the goods produced and the processes employed, but having also, and equally, a practical knowledge of how to observe, record, analyze and compare essential facts in relation to wages, supplies, expense accounts, and all else that enters into or affects the economy of production and the cost of the product."

The following paper attempts to sum up the results of modern experience in these matters, and to state the views now generally held concerning the fundamental principles which underlie industrial accounting. The conclusions reached are embodied in the Axioms; the arguments on which the latter are based are contained in the intermediate text.

AXIOMS CONCERNING MANUFACTURING COSTS

BY HENRY R. TOWNE, NEW YORK

Member of the Society

Three factors enter into the cost of each and every article of manufacture; namely, *materials, labor* and *expenses.* These constitute a tripod, a three-legged stool, which cannot stand if one of these legs be omitted. They may, and do, vary in dimension, but all three are invariably present, and a "cost" which omits any one of them is incomplete and fallacious. The formula is

$$L+M+E=C$$

in which L represents labor, M materials, E expenses and C cost. In this primary division the item "labor" includes all labor entering directly into the product, the item "material" all material entering directly into the product, and the item "expenses" (often called overhead charges, or simply overheads) all other labor, material and expenditures of every kind whatsoever.

Axiom 1 Every cost includes three fundamental factors: labor, material, expenses.

2 In most cases, however, the expenses, or overheads, divide naturally into two groups:

a Manufacturing expenses, those incident to the operation of the factory or mill; that is, those incurred in utilizing productive labor and material, and in bringing the product up to the point where it is ready to be sold.

b Commercial expenses, those incident to the commercial department of the business, including administration, salesmen, advertising, office expenses, etc.; that is, those incurred in distributing and selling the finished product.

It is highly expedient that these two should be segregated, so that each may be studied separately.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, 29 West 39th Street, New York. All papers are subject to revision.

Axiom 2 The expense factor should be split into two parts: manufacturing, commercial.

3 Letting the symbol Me represent the former, and the symbol Ce the latter,

$$L+M+Me+Ce=C$$

But a more convenient and indicative form of presenting these elementary facts, one which the writer has used for many years, is the following:

L = productive labor
$\underline{M}$ = productive material
PC = *prime cost*
$\underline{Me}$ = manufacturing expenses
SC = *shop cost*
$\underline{Ce}$ = commercial expenses
AC = *actual cost*

4 If preferred, the foregoing facts may be expressed by the following formulae:

$L+M=PC$ or *prime cost*
$PC+Me=SC$ or *shop cost*
$SC+Ce=AC$ or *actual cost*

Axiom 3 A manufacturing cost has three phases: prime cost, shop cost, actual cost.

5 On the appreciation and intelligent use of these facts hang all the laws of good business and the profits, for no business can long be operated successfully without a correct knowledge of costs, nor can that be had without a clear grasp of fundamental principles. The competitor most to be feared, while he lasts, is one who does not know his costs, nor understand how to obtain them.

Axiom 4 Accurate cost information is vital to good management.

6 Simple as are these elementary principles, their correct application in each given case is difficult, and calls for great care and intelligence. To draw correctly the line between productive and non-productive labor and material, through each of the successive stages of a productive industry, requires the combined skill of the expert manufacturer and the expert industrial accountant; the former knowing accurately all the details of the manufacturing or productive processes, and the latter knowing equally the proper methods of combining and using the recorded facts to yield the desired information. For example, what constitutes productive

labor? In the case of a machinist operating a lathe clearly it includes his wages while his lathe is turning out product and also while it is standing still during the time he is dressing the tool to do the work properly. But if the tool-dressing is done for him, as it is under good modern practice, how shall the time and wages of the tool-dresser be classified? So as to productive material, shall the tool, the file, the waste, the oil, which are consumed or used up in making the product be classified as productive or as non-productive material? The answers to these questions depend on the surrounding facts in each case, and are as varied as the cases are infinite in number and variety. The writer is not attempting here to answer such questions, but merely to point out and emphasize underlying principles. This much is clear, that every individual item of expenditure, large and small, must ultimately classify under one of the three great heads above referred to, labor, materials or expenses, and that profit, or loss, is the difference between actual cost and the net price realized.

Axiom 5 Accurate costs imply the correct classification of every expenditure.

7 The distribution of actual costs among these heads, or, preferably, into the four groups or divisions indicated above, varies widely in different industries and with different products. This is illustrated by the following table, relating to four distinct lines of actual product, in which the several elements have been reduced to terms of the actual cost of the product when finished and sold.

Nos.	1	2	3	4
L = productive labor........	28	17	29	19
M = productive material......	38	33	25	37
PC = prime cost..........	66	50	54	56
Me = manufacturing expenses..	24	20	28	22
SC = shop cost...........	90	70	82	78
Ce = commercial expenses.....	10	30	18	22
AC = actual cost.........	100	100	100	100
Total expenses (that is, $Me+Ce$) =	34	50	46	44

8 The figures in the above table illustrate the hopeless state of mind of manufacturers, some of whom still survive, who delude themselves by the belief that the sum of labor and materials (prime cost) represents the actual cost of the product, and that the difference between that and the selling price is profit. They show, on the contrary, that, in the four examples to which the figures relate, the

prime cost constitutes only from one-half to two-thirds of the actual cost, and that the expenses, or overheads, incident to conducting the business and marketing the product, contribute from one-third to one-half of the total or actual cost. It seems probable that, if the facts concerning all manufacturing industries could be ascertained and averaged, the "three-legged stool" would be found to stand nearly level, its three legs being approximately of equal length, although differing widely in individual cases.

9 All cost accounting should aim to segregate charges wherever this can be done accurately. Thus the major part of the items constituting productive labor and material can and should be charged directly to their respective accounts, L and M.

> *Axiom 6 Every productive expenditure should be charged directly to its proper account.*

10 All other items, however, which cannot be so segregated must be aggregated into one or several groups and their total apportioned among the proper accounts on some carefully determined but necessarily arbitrary basis.

> *Axiom 7 All non-productive expenditures should be properly grouped for final distribution.*

11 Manufacturing expenses may be apportioned as a ratio or percentage of labor L, of material M, or of labor and material $L+M$. The usual bases are either L or $L+M$. The writer believes that in most cases the closest conformity to actual facts will be attained by distributing manufacturing expenses in the ratio of productive labor, hand and machine, because usually the volume of indirect expenses of works operation will be far more influenced by the pay-roll, that is, by the number and kinds of employees, than by the materials bill, that is, the amount paid for the material of production. Moreover, the former is relatively stable, while the latter fluctuates with market changes. Therefore it is advisable that Me should be a function of L, that manufacturing expenses should be apportioned as a percentage of productive labor, although in some cases they may properly be apportioned per machine, or per unit of floor space.

> *Axiom 8 The normal basis for distributing manufacturing expenses is productive labor.*

12 Commercial expenses may also be apportioned as a percentage of L, of M, or of $L+M$, and frequently are, but more properly they should be apportioned on the basis of shop costs, $L+M+Me$. The reasons for this are conclusive. Production and selling are two sepa-

rate and distinct processes. The former brings the product to the point where it is completed and ready for sale; the latter then takes it over and effects the sale. The expenses incurred in each process are for its use only, and have no natural relation to the needs and uses of the other. To illustrate this, take the case of a manufacturer of cotton cloth who sells his entire product through a commission house or broker. Clearly his whole commercial expense account is covered by the commission he pays to his selling agent, and this bears a definite ratio to his shop cost, that is, to the cost of his product ready for sale. Now, take the case of another manufacturer of cotton cloth, who maintains his own selling organization and through it distributes his product. Clearly his commercial expenses offset the commission paid by his competitor, and equally bear a definite ratio to his shop cost. Both are most accurately stated and apportioned as a percentage of the shop cost, the cost of the product ready for sale. Therefore commercial expenses Ce should be distributed as a percentage of shop cost, $L+M+Me$.

Axiom 9 The normal basis for distributing commercial expenses is shop cost.

13 When the product is simple and homogeneous, for example, such as pig iron or cotton cloth, one account may suffice for all manufacturing expenses and one other for all commercial expenses, but when it is diverse or complex each of these should be sub-divided into one for each department or for each distinct class of product. In effect such a business is an aggregate of several businesses, some of which may yield better results than others, or may fluctuate more widely, and a proper accounting system should show the results of each sub-division or department separately, as well as the combined result of all. Hence arises in many cases great complexity in cost accounting, and corresponding need and opportunity for the skilled industrial accountant.

Axiom 10 An accounting system should show results both by departments and by totals.

14 In some cases the entire product consists of a single staple article, or group of articles, for which there is a constant demand and, at some price, a sure sale, such as pig iron, window glass, cotton cloth, etc. In other cases the product must conform to the specifications of the customer, and therefore cannot be made up in advance of orders, as in shipbuilding, carbuilding and the construction of buildings. The former is commonly designated as a

stock product, and the latter as contract work. The difference between these may be expressed as follows viz.:

A stock product is one which is made first and sold afterwards.

A contract product is one which is sold first and made afterwards.

Cost accounting is usually more complex and difficult in the case of contract work than in that of a staple or stock product.

Axiom 11 A contract product may require a more complex accounting system than a stock product for the accurate determination of costs.

15 The expenses of general administration overlap the manufacturing and the commercial divisions of an industrial business. Many items can and should be definitely charged to one or the other. Others may arbitrarily be apportioned between them; as, for example, the salary of an official who devotes say 70 per cent of his time to one and 30 per cent to the other. All others must be aggregated into groups for distribution by the methods adopted, as above, for distributing such expenses; as, for example, by percentages of productive labor or of shop costs. Expenditures of this kind are infinite in size, kind and number, and call for great skill and good judgment in their classification, which should be determined in advance by a clearly defined code, not left for haphazard decision by subordinates. Such a code, based on intimate knowledge of the business, on a clear perception of the information of the code is designed to yield, and on sound accounting principles, is an indispensable prerequisite to the accurate determination of costs, and equally to the intelligent conduct of any manufacturing business.

Axiom 12 An accounting system should be embodied in a code of instructions, for the guidance of those responsible for its operation.

16 For best convenience a code should provide symbols to represent the various accounts and their many combinations. For this purpose the writer for many years has used a system of letters and numbers which possesses great convenience. Letters are used to designate important departments and accounts, the significance of each letter depending on its place, as in decimal notation, in the symbol. Thus, the first letter may indicate a department, the second a sub-division of it, and the third a room or smaller unit. Stated numbers are used to indicate accounts relating to expenses of the

various kinds or groups. Such a symbol is shown by the following example, viz.:

BAC 10

in which *B* represents the department, or the class of product, against which the item is to be charged; *A* the shop in which the work is done; *C* the job, or machine, by which it is done; and 10 the kind of expense to which the charge relates, such as repairs of the machine, foreman's wages, etc. These symbols, and an explanation of their meaning and use are printed in a small book of pocket size, copies of which are furnished to all concerned. In this way a correct classification of every charge is made at the time of original entry, after which tabulation and aggregation of original charges follow automatically in accordance with the predetermined plan.

Axiom 13 Symbols are better than titles for recording charges in an extensive accounting system.

17 In any business certain expenses or losses occur from time to time which are unusual or abnormal. These may be termed "extraordinary expenses," and require special consideration. As examples of these may be cited a serious loss by fire, a curtailment of product by a strike, an abnormal loss through bad debts, an increase or decrease in value of land, etc. The loss, or profit, thus arising must of course be covered into the treasury, but this may better be done through a debit or credit to the surplus account than through a charge to the profit and loss account of the current year, for the latter plan would distort the statistical record of the year by including in it items not common to normal years. The best plan is to charge to the account of each year only the items which are normal, and to charge those which are abnormal to the surplus account. The proper purpose of the annual account is twofold, (*a*) to show the results of the year's operations, and (*b*) to contrast these results with those of preceding and succeeding years. On the other hand, all extraordinary gains or losses must be accounted for, and this may best be done through the surplus account. In this way both purposes are accomplished.

Axiom 14 Extraordinary gains or losses, in order not to distort the statistical value of the annual profit and loss record, should be covered into the surplus account between the closing of the books for the old year and the opening of the books for the new year.

18 Interest on borrowed capital is a distribution of profits, not

an expense, although often erroneously treated as the latter. To illustrate this suppose the case of two manufacturers, *A* and *B*, each having $200,000 invested in his business and each realizing 10 per cent, or $20,000, net profit available for dividends on a year's business, all of *A*'s capital being contributed in cash, while *B* has only $100,000 of cash capital, and another $100,000 of borrowed capital, on which he pays 5 per cent interest. At the close of the year *A* is in position to pay $20,000 in dividends to his stockholders, a 10 per cent return on their investment, but *B*, after paying $5000 as interest, is in position to pay $15,000 in dividends to his stockholders, a 15 per cent return on their investment. Evidently the actual profits from the operations of the year are the same in each case, only the ownership of the capital invested and the distribution of the profits being different. The accounting system should show the actual profit realized, regardless of its distribution to the owners of the capital invested in the business. On the other hand it is expedient that interest on temporary loans, and on time purchases if availed of, rebates and discounts of customers' notes, should be treated as current expenses, normal to the conduct of the business. In like manner discounts earned by cash payments should be treated collectively as part of the current earnings of the year, or else be covered into the net costs of purchases.

> *Axiom 15 Interest on borrowed capital should not be treated as an operating expense, but should be charged direct to the profit and loss account of the year.*

19 Interest on all capital invested in a business may or may not be deducted before stating the final profits of the year. Here no principle is involved, but merely convention or individual preference. Usage, however, has practically determined that it shall not be deducted; that the final, or net, profit should indicate the return on capital, the amount which capital has earned. Stated thus it can readily be compared with what the same capital would earn if invested otherwise; as, for example, in government or railroad bonds, in mortgages, etc. If interest is deducted at all, as is done, for example, under some profit-sharing plans before allotting anything to the beneficiaries of the plan, it should be computed on the total capital invested in the business, including therein the surplus account; that is, surplus profits of previous years retained in the business and invested in plant or merchandise.

> *Axiom 16 Final profits properly signify the amount earned by*

the capital invested. If interest on capital is deducted this fact should be stated, and interest should be computed on the total capital employed.

20 Where a business is divided into several or many departments it is very desirable that the accounting system should show the profits or earnings of each of them separately, and this is usually feasible, except as to annual depreciations and as to interest charges. In some cases either or both of these items can accurately be distributed among the several departments, and if so, they should be so treated. Where they cannot be so distributed they should be deducted in a lump from the sum of departmental profits, and in this case it becomes convenient to adopt terms to designate clearly the profit account at its various stages. For this purpose the writer has found the following terms satisfactory:

Gross profits: the aggregate profits of all departments, prior to deducting depreciations and interest.

Net earnings: the gross profits after deducting depreciations.

Net profits: the net earnings after deducting interest on borrowed capital.

21 In comparing the results realized in two or more comparable concerns or businesses it is essential to contrast profits at the same stage in each case, and to employ terms which are mutually understood as to their precise meanings. No standard as to these terms has yet been established. The proper basis of comparison usually is that indicated above by the term net earnings, which eliminates the variations due to the employment or non-employment of borrowed capital.

Axiom 17 Terms used to designate profits should indicate clearly the stage of profits to which they refer, and should be mutually understood.

22 Inventory valuations are an important factor in determining profits. Usually an actual inventory is taken only once a year. The merchandise inventory includes raw materials, stock in process, finished goods, and general supplies. A standard basis of valuation for each of these groups should be adopted and maintained from year to year. Raw materials, such as pig iron, raw sugar, baled cotton, ingot copper, etc., are often subject to wide fluctuations in market values or costs, and the question thus arises as to the proper inventory valuation of them, whether at cost, at market value at date of inventory, or on some arbitrary basis. If the effect of such

fluctuations is negligible, that is, is small in ratio to the annual total of the account, either of the first two methods above stated may be used. If the fluctuations are large, however, either in range or in their effect on the annual total, that is, if they materially influence the profit and loss account of the year, some arbitrary plan of accounting for them should be adopted. In devising this, the twofold purpose of the annual account, the operative and the statistical, should be kept in view. If the effect of the fluctuations is moderate in its ratio to the annual account a sound method consists in taking the mean, either of market prices or of actual purchase prices for say three or five years, as the basis for inventory valuations, and also for use in the compilation of costs, thus conforming to the average trend of market values but avoiding frequent and temporary changes. If, however, the effect of these fluctuations is serious or vital in determining the results of the business, a new factor is brought into the accounting problem, namely that of trading or speculating on the market. In the case of a sugar refinery or a cotton mill, for example, large profits or losses may result from market changes in the price of raw sugar or of baled cotton, or from the operations of the purchasing department. Obviously such gains and losses are totally unrelated to the economy and efficiency of the productive department, and to include them in its accounting might so distort it as to destroy its usefulness and its statistical value. In such cases a separate trading account should be established, through which to ascertain the profit or loss of the year in operating on the market for the raw material, the latter being charged to the manufacturing department at a constant price, conformed from time to time to average market conditions, this price being used also for inventory and cost purposes. In other words, the results of speculation on the market, however legitimate or necessary, should be segregated from the results of the normal operations of the plant.

Axiom 18 Speculative profits and losses should be segregated from those due to the normal operations of a business.

23 The inventory valuation of stock in process, that is, of stock in a partly manufactured condition, should be such as to cover the prime cost of the material, and of the productive labor already expended upon it, plus a ratable charge for manufacturing expenses. The inventory valuation of finished stock, that is, of stock completed ready for sale, should be on the basis of shop cost, not of actual cost, because the latter includes the cost of selling, and this has not

yet been incurred. A paradox, apparent but not real, is created when the cost of a product is substantially reduced, because thereby the inventory value, and therefore the profit of the year, is reduced. If the inventory value at the beginning of the year were $1000, and if during the year the cost were reduced 10 per cent, obviously, if the quantity on hand at the close of the year were the same, the inventory value would be $900, thus showing a shrinkage of $100. In the following year, however, this apparent loss would be converted into an actual profit.

> *Axiom 19 A reduction in cost implies a corresponding reduction in inventory.*

24 The annual inventory may properly include as assets certain items previously classified as expenses. One example of this kind is the premium on unexpired insurance. Another is the cost of a trade catalogue intended to serve say for five years. To charge the whole of important expenditures of this kind into the current expense account of the year in which they are incurred would tend to distort its statistical accuracy, and hence would be bad accounting. The proper treatment of such expenses is to determine the period they apply to, and to charge off a proportionate part in each month or year during that period, carrying the remainder in the inventory.

> *Axiom 20 Expenditures in one year which cover the requirements of several years should be distributed over the years to which they fairly apply.*

25 The inventory valuation of all property other than merchandise should be on the basis of its fair value in the business as that of a going concern, which usually is the cost to replace, with due allowance for wear and tear. An annual inventory of all property, by actual enumeration and count, is indispensable to the proper conduct of any manufacturing business, and in some cases more frequent inventories of the merchandise stock are expedient. Without such annual inventories no determination of annual results is reliable or of much value.

> *Axiom 21 An annual inventory of all property is indispensable to accurate knowledge and to good management.*

26 The question of depreciation of fixed property enters into all industrial accounting, and should be treated in connection with the inventory. In this, as in all discretionary matters of accounting, the aim should be to find and follow the median line, the mean between ultra-conservatism and radicalism. All fixed property, ex-

cepting land, depreciates and tends to become obsolete. Normal repairs and maintenance should of course be charged to current operating expenses, not added to cost or value, and these should fairly be considered in fixing the ratio of depreciation. Where a building or a machine is maintained in perfect condition obviously it depreciates more slowly than one which is neglected. A building may be so maintained as to depreciate little or not at all. The proper rate of depreciation for each class or kind of fixed property is a matter of good judgment, for which no rules can be laid down. It may be as low as 1 per cent per annum, and in exceptional cases may be as high as 20 per cent. Usually it ranges from 2½ to 10 per cent. When profits are abnormally large the allowance for depreciation may wisely be larger than when they are merely normal, but the normal allowance should be made even when no profit is realized. Under average conditions it usually ranges between 10 and 15 per cent of the annual profits. A revaluation of all fixed property by outside experts or appraisers, at intervals of five or ten years, is expedient and usually worth its cost. Abnormal increases or decreases in the value of such property, as for example an increase in the value of land, or the loss due to the demolition of an obsolete building, should be covered into the surplus account, not into the profit and loss account of the year.

Axiom 22 Valuations of fixed property should be subject to annual review and to fair depreciation.

27 Finally, the aim and object of every accounting and cost system should be to afford true and accurate information as to facts. It is based on facts; it should embody and present facts, and naught else. To exaggerate facts and to show fictitious profits and values, is no worse than to depreciate facts and to conceal true profits and values.

Axiom 23 An accounting system should present facts, without bias in any direction.

28 Accounting, in its application to general business affairs, has long been a highly developed science, but is comparatively a new one in its specialized application to modern industry, with its vast and complex development. The creation of a *Correct Science of Industrial Accounting* and costs should be the desire and aim of all who are concerned with industrial management. To accomplish this, three things at least are needed:

a Clear understanding of fundamental principles *b* Definite terminology, generally understood and accepted

c Free interchange of the data of practice, whereby the adoption of sound principles may be promoted, the experience of each may be available to all, the best methods may become established, and, above all, a standard system may ultimately be created.

29 The accomplishment of these results, by affording complete and accurate knowledge of the essential facts pertaining to industrial efficiency, and to the costs of production, will tend greatly and permanently to promote the development of American industry, and to aid it in securing its full share of the markets of the world.

THE POWER PLANTS OF TEXTILE MILLS

By John A. Stevens

ABSTRACT OF PAPER

This paper describes the purposes which should be served by power plants in textile mills.

It calls attention to the individuality of each plant and to the special design and adaptation necessary if it is best to serve its purpose, showing how manufacturing conditions as regards use of low-pressure steam in textile mills make well designed plants for such mills differ from those which would ordinarily be best and most economical for other lines of industry.

The degree to which power plants in textile mills are complicated by use of water power requiring relaying by steam units is discussed, and the effectiveness of electric transmission for securing necessary flexibility under these conditions pointed out.

Several power stations recently installed under the writer's supervision are described and illustrated, the purpose being to call attention to particular features of each plant which illustrate points mentioned in the paper.

A more careful consideration of the advantages which might be secured by groups of mill owners combining to produce power in a single station jointly owned and operated by the group is urged.

THE POWER PLANTS OF TEXTILE MILLS

By John A. Stevens, Lowell, Mass.

Member of the Society

The purpose of this paper is to place anew before those engaged in managing and operating textile mills some of the salient points which have to be considered in providing a power plant which shall do its best work in connection with such mills; and to show how the differing conditions make each plant a problem worthy of special consideration.

2 The power plant is apt to be regarded merely as an expense item and although this is only a small portion of the total of such items, usually not more than 3 to 5 per cent, it is so large a sum in dollars per year that possible savings therefrom possess a real interest for managers of textile plants and are receiving serious consideration whenever they can be properly presented.

DIRECT AND INDIRECT SAVINGS

3 The possible savings in connection with a textile power plant are at least of two kinds: (*a*) The direct savings brought about by producing through efficient apparatus a given quantity of needed power for a reduced expenditure of money after all expenses chargeable to power have been considered; and (*b*) the incidental savings realized through installing new equipment carefully arranged to meet the manufacturing needs of the plant, thus bringing about increased and more convenient and bettered production, or a decreased cost of manufacture. These frequently are of greater importance than the direct savings.

SPECIAL ADAPTATION REQUIRED

4 It is a fact so self-evident as hardly to need stating that in order to serve most perfectly all the requirements of a given plant and thus take advantage of all possible savings, its power

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equipment must be specially adapted to the individual plant, the conditions and requirements of which are never twice alike.

5 Textile plants, of course, can be roughly grouped into classes depending upon the kind of goods which they produce, whether worsteds and woolens, cottons, linens or silks; whether they produce yarns or cloth; whether their product is plain or colored; to what degree this product is finished and by what processes, that is, whether there are dye houses and bleach houses as part of the equipment, requiring large quantities of manufacturing steam.

6 Certain features of their power house design will probably depend, to a considerable extent, upon which of these groups they belong to, but the fact that there cannot be a standard design for a power plant, for any purpose, whether for public service, textile mill or other use, furnishes the real reason for the existence of the power plant engineer.

CONDITIONS MET WITH IN CENTRAL STATIONS

7 While the conditions and elements which make for diversity in design or most desirable equipment of central stations for public service corporations are everywhere admitted to be great enough to necessitate the careful working out of all features by a competent group of engineers, such equipment, when of small size, is generally much more capable of standardization than is that of the textile plant. In the great majority of cases the public service station has to do solely with the furnishing of electric energy for use by its customers in the form of power and lighting. It has, of course, no use for low-pressure steam in considerable quantities and its requirements for heating steam are insignificant.

8 In fact, so far as the question of low-pressure steam enters at all into its scheme of operation, it is rather one of avoidance than otherwise. Its problem is to see that no more is produced at any time than can be made use of for the purpose of heating feedwater, and anything beyond this quantity is an absolute waste.

9 The influence of this condition extends so far that not infrequently it has led such plants to forego the savings from an economizer, because if such a unit were used economically, there would be so little remaining use for exhaust steam that auxiliaries, which for reasons of dependability and flexibility

it has seemed best to keep as non-condensing steam units, would be exhausting to atmosphere.

10 The public service station manager is thus prevented from the possibility of realizing in his own plant large savings from conditions which can be obtained by the manager of the textile plant, savings, too, which must be thus taken advantage of if the item of total coal cost in that plant is to be kept at minimum.

11 This absolute dissimilarity of conditions goes far to explain why it has sometimes happened that engineers and power sales agents who have received their training mainly in situations where there was no possible use for low-pressure steam, have utterly failed to realize the extent of savings which may accompany such use. They have sometimes gone so far as to cause the abandoning of steam-using machinery and the purchase of current for power, even in plants containing dye houses, with the result that the total bill for power and manufacturing steam was thereby increased very nearly to the extent of the power bill. This, of course, represents an extreme case, and is mentioned here merely as a means of bringing out clearly the extent to which conditions may sometimes influence the proper solution of a textile power plant problem.

COMBINED POWER AND MANUFACTURING STEAM USING UNITS

12 The plant which can obtain power from units properly combined with the supplying of steam for necessary manufacturing or heating processes has open to it possibilities for securing the same at costs far below those available to others not thus fortunately circumstanced.

13 The reasons why, when conditions are suitable, such considerable savings can be realized by making the power producing unit practically a reducing valve for the manufacturing steam line, or looked at from the other point of view taking the manufacturing steam from the power line as a sort of byproduct, are well understood by all engineers who have had occasion to consider the matter.

14 Although the practice is old, dating back many years, we still find men occasionally who feel that the idea of getting the work required from an engine and from a dye tub with less coal when the latter takes its steam through the engine than when both are supplied by separate lines from the boiler is a

sort of perpetual motion scheme—a producing of something from nothing.

15 They fail to take into consideration that there is a tremendous amount of heat energy rejected through the condenser even by the most efficient engine or turbine, while the dye tub is particularly well fitted to make use of this very energy which the engine or turbine cannot use. On the other hand, the dye tub cannot as efficiently utilize the energy from the high-pressure end of the range and from which, in the combined unit, power is obtained.

16 When the steam turbine first began to replace the steam engine, it generally was not designed to meet this condition of furnishing low-pressure steam and the fact that it was better able to use steam efficiently through the high vacuum ranges doubtless led to the feeling that it was not so well adapted as the steam engine for supplying steam for manufacturing purposes.

17 Later experience has shown the fallacy of such belief and now turbines are designed with special reference to such uses. It is also worth noting that at any given pressure steam leaves the turbine considerably dryer than it leaves the engine.

18 The quantity of manufacturing steam required and the regularity of demand for the same varies greatly in different mills. For best results individual conditions should receive careful study before equipment is chosen. Where demands are large and constant non-condensing units may offer the best solution. Where demands are extremely variable some system of bleeding a portion of the steam from the receiver or stage of a condensing unit is more practicable.

USE OF CONDENSING WATER DISCHARGE IN TEXTILE PLANTS

19 It sometimes happens that power producing and manufacturing machinery can be very advantageously combined in another way. When there is a large and constant demand for warm water to be used by textile machines, the cooling water from condensers, carrying with it the rejected and otherwise waste heat from steam, can be taken to these manufacturing uses where, unless this were done, cold water would have to be used and heated by steam for which coal would be especially burned. Thus put to work the condensing water may save a portion or all of the above-mentioned coal, depending upon the temperature of water required.

20 The drawbacks to this scheme which prevent its more frequent use are that the demand is apt to be for a relatively small amount of water at a considerably high temperature, while the supply from a normal and efficient working condenser would be that of a larger quantity of water at a temperature only moderately in excess of that of the cold water supply. Again the steam engine, which can better stand a rather low vacuum than can the steam turbine, under this condition requires the use of a surface condenser in order to avoid the presence of even a small quantity of oil which usually is objectionable.

CONDITIONS COMPLICATED BY USE OF WATER POWER

21 It is believed that the power plant situation of a textile mill is, more frequently than with any other type of plant, complicated by the presence of water power equipment developed to such an extent that a considerable portion of it requires relaying with some form of coal consuming units. This is especially true in those sections where the textile industry had its beginning and earlier growth, and has come about naturally from the fact that the textile industries are among the oldest of considerable importance in the country. There are many great textile corporations whose history goes well back towards the beginning of the last century, or at least to a time when water privileges could be more readily obtained than is the case today, and when steam power was but little developed and relatively inefficient. Water power developments of great extent and cost were made during those early days, the rights in which are still held by the original textile companies or by their successors, and which practically make financially necessary and good policy the continued use of water power to an extent which would be prohibitive were the developments being made from the beginning today.

22 Other industries and especially public service corporations, being of later growth, missed the opportunity to share in the rights of these early water power developments; or when they secured water privileges, have in the main because of expense, developed them less fully. Their equipment in any event is more likely to be modern and thus fit in better with present-day power-generating machinery.

THE NEW POWER PLANT DEVELOPMENT

23 The problems of textile power plants can be subdivided into two classes: (*a*) entirely new developments, and (*b*) those which have to do with changes or further developments in an existing plant.

24 When an entirely new power plant is under consideration, the fundamental points are the same whether it be for a textile mill or any other purpose. These are accessibility to railroad or barge for handling coal and ashes, coal storage, boiler feed and condensing water supply, also convenience of and ample space for future expansion of plant.

25 *Importance of Track Facilities.* The importance of laying out a new manufacturing plant or making additions to an old one to secure the best, or at least workable, track facilities cannot be too strongly emphasized. The point is so old and so fundamental, so well-known to all engineers and plant managers, that the writer would hesitate to mention it had he not found in actual practice that it is frequently overlooked or given too little consideration.

26 With the rapid increase in size and weight of rolling stock, the railroads of the country have found it necessary to increase greatly the minimum radius of curves over which they will undertake to handle cars, with the result that where the small cars of a decade ago got around with no serious inconvenience, the large equipment of today too often can not go. The manufacturer who builds an important mill group without first having his track connections definitely laid out to the approval of railroad engineers, is apt to find that intended connections are impossible, or at least awkward and unduly expensive.

27 *Providing Room for Future Development.* The importance of locating a new power house to have ample room for growth, is another point which ought not to require mention, but the fact that rapidly growing plants are so frequently found with this important portion of their equipment needlessly built into a hole, seems to justify some mention of the subject.

28 Sometimes such building is forced upon a concern by the circumstances of location and business conditions, and it is then of course justifiable as it has been done with full recognition of the fact that it is going to prove expensive later on. The

greater portion, however, of such building seems to have come about through a lack of conception that there were possibilities of greater things in the future and through an unwisely directed effort to economize in the first cost of power equipment, an effort which not infrequently has failed in itself since as much has been spent in the location impossible of further development as would have been required to meet the existing demands and still provide the possibility of further needed growth.

29 The serious feature of building a power plant in a cramped location lies in the fact that if the growth of the manufacturing is such that more power is required, it is almost impossible to remedy the condition. The existing plant represents a heavy investment in equipment which should be good for many years more service. To move it to a location where expansion can take place, or to abandon it and substitute new equipment means, in the latter case, a fixed cost for the new station well nigh or absolutely prohibitive; to build another independent equipment to meet new demands and let the old remain, means continuous operating expenses much higher than should be secured with a single station. The providing of ample facilities for growth if undertaken at the proper time is usually so simple and comparatively inexpensive that this feature should never be neglected.

30 *Location with Relation to Future Manufacturing Growth.* Another point frequently overlooked in locating a power house is that it shall not be in the way of future development of the manufacturing buildings. It often happens that some department must be enlarged, but the power plant is in the way. As the latter usually cannot be moved, the addition to the manufacturing department must be put somewhere else, disrupting the organization of the factory and breaking the proper passage of the product from raw stock to finished goods. This results in increased handling and trucking, and increases the cost of the finished product.

OLD POWER PLANT DEVELOPMENT REQUIRING ADDITIONS

31 The subject of the second subdivisions, viz., problems which have to do with changes or further development in an existing plant, of necessity make up the greater portion of the work of a power plant engineer.

32 *Power Plant Analysis a Method of Handling Preliminary*

Work. The method of attack, which, in the writer's own experience, has proved most satisfactory in getting at the requirements of these individual problems and so presenting them as to lead to well advised action on the part of plant managers, is by means of the plant analysis, a detailed study of and report upon that mill's whole power plant situation.

33 Through the gathering of data necessary for such a report, the engineers in charge of the work become better acquainted than in any other way with the resources, the possibilities and the requirements of that plant.

34 Such a report aims to show accurately the extent and value of present equipment; how much and where energy is produced and how distributed, with details as to quantities required for such subdivisions as frictional resistance, useful power and lighting, and steam and coal consumed for manufacturing and heating; also the cost of producing power with present equipment. These costs are compared with those which should result through installing other equipment in accordance with one of several schemes, preliminary plans and estimates for which are worked out. Such schemes are thus compared, the strong and weak points of each developed and as far as possible the whole question is reduced to a financial basis looking either towards the greatest return from the least new expenditure of money, or the greatest return when service-ability through a period of years and adaptability to plant conditions and growth are considered.

35 *Power Plant in Old Mill an Adaptation to Existing Conditions.* The power plant problem in an old textile mill is always one of adaptation not only to the requirements for power in all its phases, but more especially, as has previously been pointed out, to the more difficult requirements of having the new equipment fit in with that already in the plant and thus form a satisfactorily working unit at least expense, while still conserving as far as possible the money already expended.

36 The special lines along which adaptation has to be secured in any individual textile plant of course depend to a great degree upon the complexity of its power plant equipment and power uses, differing all the way between the comparatively simple condition of an all steam using plant burning coal for power uses only, and the plant making large use both of manu-

facturing steam taken from power producing units and of water wheel equipment requiring extensive relaying by steam.

37 The question of power plant location will of course be affected by the presence of one or all of these factors, for the grouping of this equipment, so that it can be handled with the fewest possible men, has a considerable effect on the final cost of power. If steam is to be used through engines or turbines for manufacturing purposes, it is very desirable that the location of the power house should be near that of these manufacturing uses, both to avoid drop of pressure in the large steam mains, and to keep the cost as low as possible.

38 *Electric Transmission as an Aid to Adaptation.* The development of electric transmission makes possible the gathering together of all steam power producing machinery in a single station under the charge of a comparatively small group of men and leaves the designer free to locate that station where it will have room for further development, be convenient for coal handling, and out of the way of manufacturing processes. It sets free for manufacturing use a considerable portion of space which was formerly taken up by belt towers and belt ways, main lineshafting, bevel gears, quarter turn and rope drives, all of which were heavy and extremely expensive both as to first cost and maintenance.

39 Above all it enables manufacturing buildings to be placed in any manner best suited to location and manufacturing conditions, and renders easy the changing and regrouping of machinery which in some departments of textile work is constantly going on owing to changed demands for production.

40 *Electric Transmission in Connection with Water Power.* By the installing of generators on waterwheels it gives this portion of power a flexibility which it has never before had and it thus somewhat simplifies the serious problem of relaying this power.

41 In many instances it is found possible to gain the necessary flexibility and prolong for a considerable time the useful life of old waterwheel equipment by means of installing belted generators connected to the main shafts of these old wheels. While this is not an efficient means of generating and transmitting power, the relatively small cost at which the necessary added equipment can frequently be installed, compared with that of an

entire new equipment of waterwheels and direct-connected generators, makes the real cost of power thus secured for the time being very attractive and retards the otherwise necessary expenditure of large sums of money.

PLANTS WHICH ILLUSTRATE THE POINTS MENTIONED

42 A brief description of several installations which have recently been made under the supervision of the writer's office serve to illustrate the working out of many of these points. Before beginning construction work on some of these plants which presented especially difficult points, the writer consulted freely with engineers who have had large experience and wishes to make acknowledgment of the courtesy and assistance received from them.

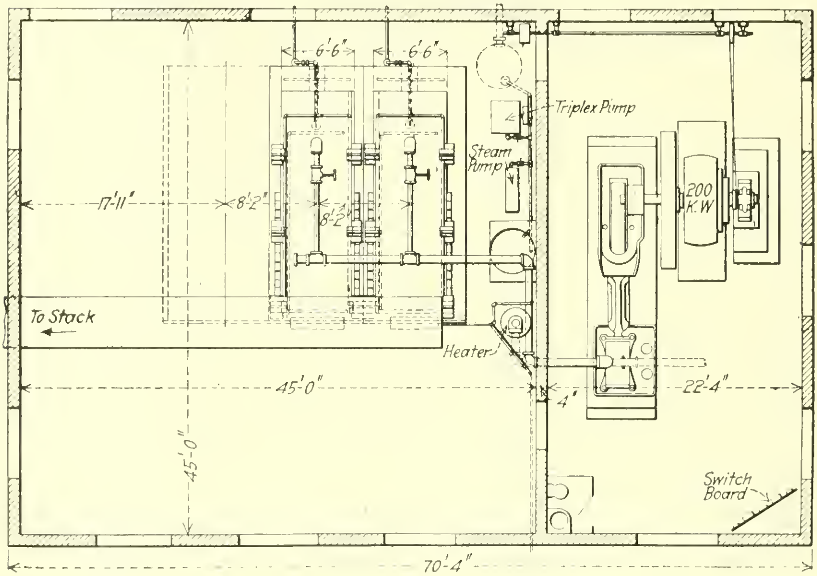

Fig. 1 Plan View of Power Plant for the Bristol Manufacturing Company

A SIMPLE DEVELOPMENT

43 Figs. 1 and 2 show the plan view and outside of the station of The Bristol Manufacturing Company, Bristol, Conn., to illustrate a small and very simple power development. This power house is wholly of monolithic reinforced concrete.

44 The station runs in conjunction with waterwheels and is situated in an irregular corner of the property between a public street and the railroad, and out of the way of valuable manufacturing space. It provides room for expansion of the power plant up to the limits of the possible development of the property.

44 The plant consists of horizontal return tubular boilers and a Cooper Corliss engine driving a direct-current generator. The power for the pumps and auxiliaries is taken direct from engine during running hours. The waterwheels are mechanically connected to mill drives and used to fullest extent possible at all times. The motors drive on to same main shafts and supply the varying make-up load.

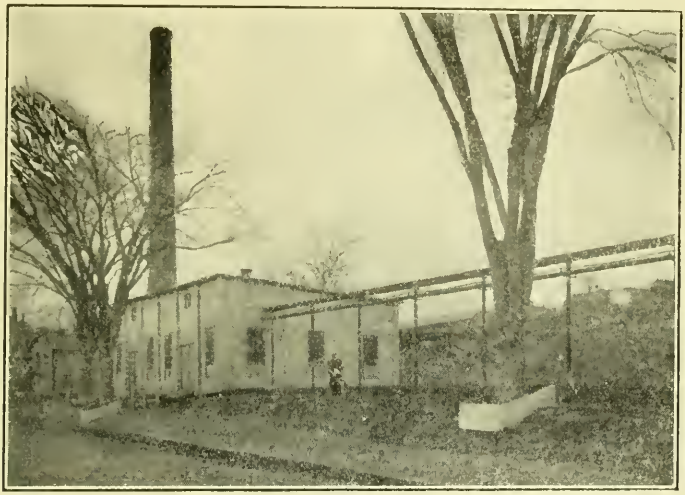

Fig. 2 Outside View of Power Plant of Bristol Manufacturing Company

45 Fig. 2 shows the simple system of supporting both manufacturing steam line and cables from wrought-iron columns in the mill yard.

A PLAIN GOODS COTTON MILL

46 An example of a simple station for a plain goods cotton mill is furnished by that of The Salmon Falls Manufacturing Company, Salmon Falls, N. H. (Figs. 3a and 3b).

47 Their need was for added steam equipment both to

furnish more power and to relay an extremely variable water power development. This was furnished by the building of a concrete and brick turbine house adjoining the old boiler house and installing a 750-kw, 3600-r.p.m. General Electric steam turbine, connected to a Westinghouse LeBlanc condenser. Arrangement was made for taking the heater make-up water for boiler-feed supply from the discharge of this condenser.

48 The turbine house is built with one temporary end, space being provided for extension both of turbine and boiler rooms.

49 The present boiler house situation was met by replacing old worn-out boilers with new Manning type boilers.

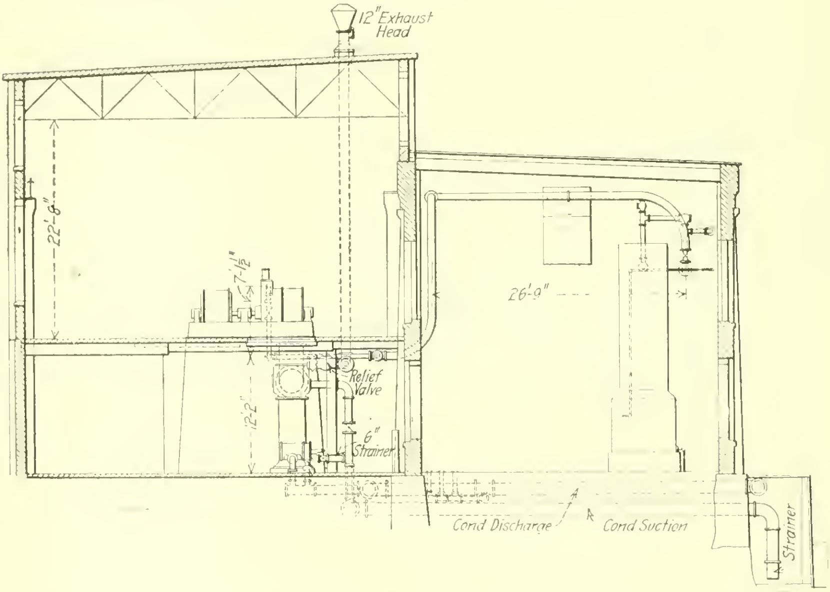

Fig. 3a Elevation of Power Plant for Salmon Falls Manufacturing Company

50 The method of tying in with the old water power development which here proved most practicable was by installing rather large sized motors connecting to the main shafting groups, and when water gave out disconnecting from waterwheel by means of clutches, many of which were a part of the old equipment.

51 As a part of this development, the entire mill group was equipped with an electric lighting system consisting of tungsten lamps. These replaced gas lighting from a gas plant which

the owners themselves operated. By this change they secured more and better illumination and a more convenient source of light at reduced cost.

A REVISION OF WATER POWER DEVELOPMENT

52 A recently installed hydroelectric unit for the Merrimack Manufacturing Company, Lowell, Mass., serves well to illustrate certain conditions which frequently have to be met in connection with water power revision in an old textile mill when such becomes necessary.

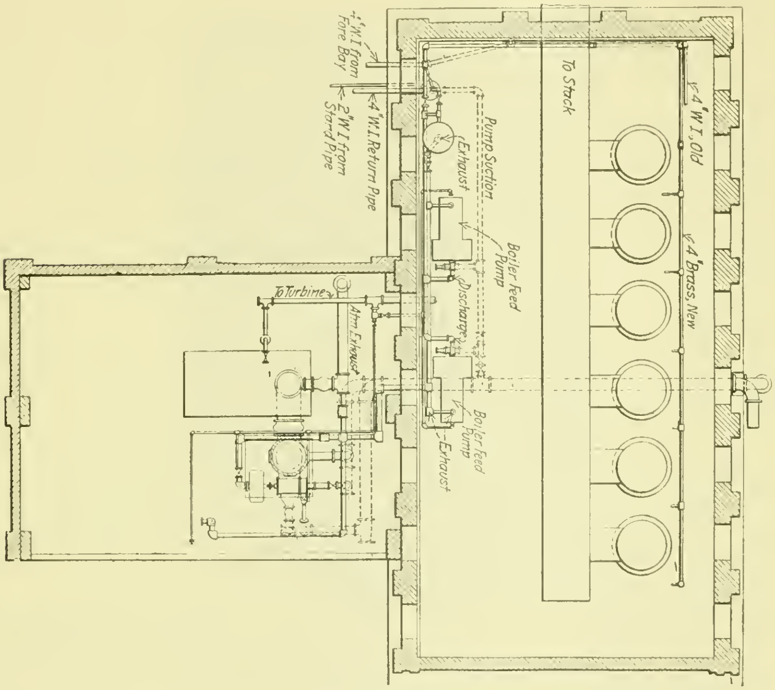

Fig. 3b Plan of Power Plant for Salmon Falls Manufacturing Company

53 A considerable portion of the power required for this group of mills had originally been furnished from Boyden wheels connected by bevel gears to systems of main lineshafts in mill basements. As is always the case with any water power development in that locality this requires extensive relaying with some form of coal burning units and had been provided by

means of steam engines geared or driving direct on to certain groups of this main shafting.

54 In order to provide power for one mill group situated on a higher level, there had been in service a line of inclined shafting some 220 ft. long driving through two sets of bevel gears from the extreme end of a main shaft about 700 ft. long, on which had been geared 5 waterwheels and to the other end of which a highly economical triple expansion vertical engine (Fig. 4) was direct-connected.

55 *Economical Direct Engine Drive.* This engine uses steam at pressures from 120 to 200 lb. per sq. in. according to load, and by repeated careful tests was found to have an economic range anywhere between 700 and 2000 i.h.p. and thus is well suited to the extreme variations in load which it has to meet, and exemplifies a simple and efficient means of relaying water power. The direct connection of engine to shaft, does away with transmission losses due to ropes or belts and the heavy up-keep on same. The engine might sometimes preferably be placed midway the length of shaft, but in this instance was placed at one end both better to conform to requirements as to available space and to make main high-pressure steam lines as short as possible.

56 *Waterwheel Equipment.* The waterwheels which had originally driven the picker building, and a portion of those on the main lineshaft, were entirely worn out, having been installed in the early 50's and in order to make use of water power required replacing by a new development.

57 Through improvements in the river bed, its average level had been lowered several feet leaving the wheel pits and tail races of these old wheels in a position where for much of the year they could not take advantage of the full available water fall.

58 There were already in service enough good wheels on good mechanical drives to absorb that portion of water power which can be depended upon at all times and does not require relaying.

59 *New Development Determined Upon.* While it is the writer's opinion that the more direct portions of these mechanical drives cannot be improved upon by any system of electric transmission, it was desirable in carrying out the new water-

wheel development to do away with the portions dependent upon much gearing and to provide for flexibility, so that this portion of power could be used where mill conditions should demand. To meet these requirements electric transmission again seemed to offer the best solution.

60 It was decided to drive the Picker Building by motor and the mill formerly driven by inclined shaft, and to make use of the main feeder and tail race of one of the old wheels so far as possible in the new development. The basement of the old mill building was too low for a suitable generator room without interfering seriously with valuable mill space, and in any event the old main lineshaft could not be disturbed.

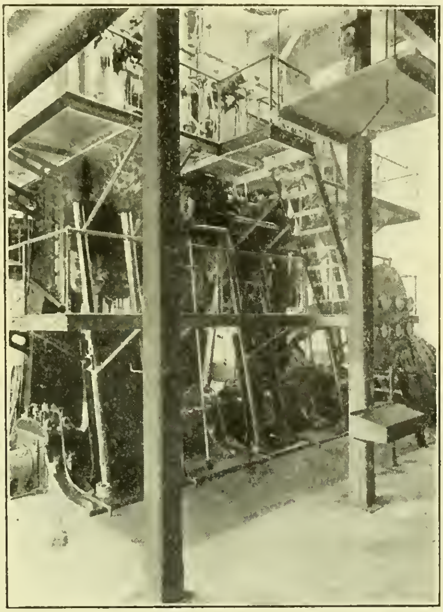

Fig. 4 Direct-Connected Triple-Expansion Steam Engine in Plant of Merrimack Manufacturing Company

61 The original feeder from a point just inside the mill was extended and gradually enlarged from a diameter of 8 ft. to 12 ft., being carried down through the side of the old wheel pit, under the main shaft, then up and out through the mill wall

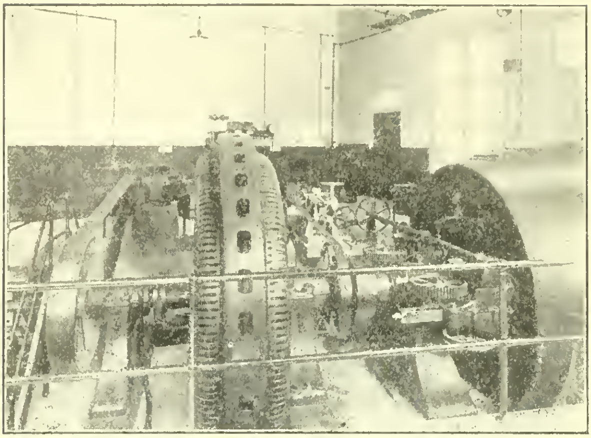

Fig. 5 Interior of Wheel Room showing Waterwheel and Generator at Merrimack Manufacturing Company

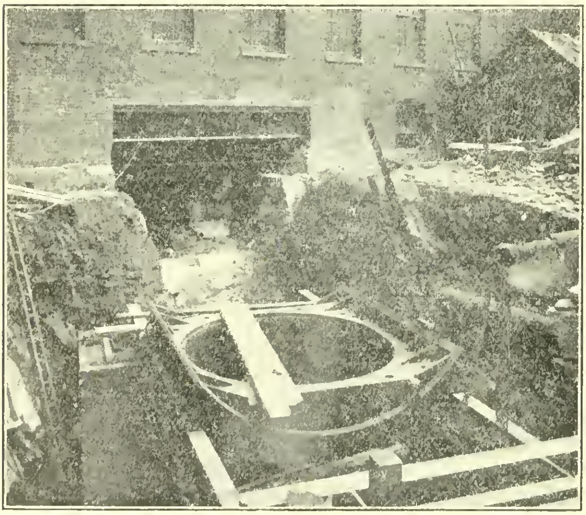

Fig. 6 View of Saddle for Draft Tube and Opening for Main Feeder Waterwheel at Merrimack Manufacturing Company

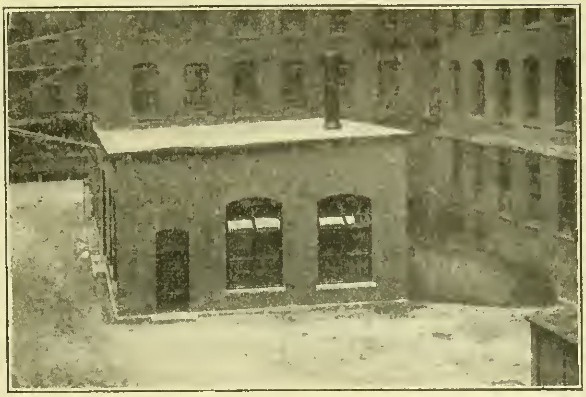

Fig. 7 Exterior View of Wheel House at Merrimack Manufacturing Company

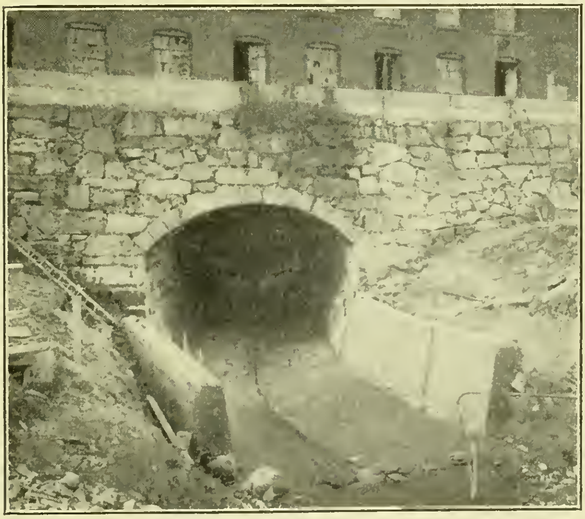

Fig. 8 View looking into Tail Race of Waterwheel at Merrimack Manufacturing Company, showing Concrete Lining in Portion newly cut down

and to the entrance of the wheel casing situated in a small independent building outside and apart from the main mill.

62 This wheel was so located as to have its draft tube discharged directly into the old tail race which was cut down for a depth of about 6 ft., its whole length, to the river. This cutting was in solid rock and the bottom and sides were made smooth with concrete (Fig. 8). In this way the full present fall from canal to river is made available and upon the test of wheel the losses through feeder and tail race were found to be small. The Holyoke Machine Company furnished the wheels which work normally under from 36 ft. to 39 ft. fall and run at 200 r.p.m. Current is furnished by a 1100 kv.a., 2300-volt 3-phase alternating current generator, and the greater portion is used on 2200-volt motors.

A STEAM TURBINE STATION FOR PLAIN GOODS COTTON MILL

63 A larger installation and one calling for considerable adaptation to local conditions is that of The Boott Mills, Lowell, Mass. This is a plain cotton goods mill doing neither bleaching nor dyeing and employing manufacturing steam only for slashing and a few minor uses.

64 A considerable portion of its power is at times furnished by nine old vertical waterwheels subdivided into four groups and located in the various mill buildings. The power thus available is not far from 3650 h.p. maximum, and 2920 normal, but decreases to between 1300 and 1400 h.p. during extreme dry weather periods, thus requiring extensive relaying. At the time when the new steam plant development was undertaken, power in addition to that of waterwheels and for the relaying of same was being furnished by two sets of old twin Corliss condensing engines and by an extensive purchase of current from a public service corporation.

65 The boiler equipment consisted of six B. & W. boilers of 305 normal h.p. each, all in good condition, and a good chimney 200 ft. high. The boilers and engines occupied the lower portion of one of the mill buildings. Serious shortage of power at times of low water made a large increase of steam equipment necessary.

66 The value of the existing boiler plant together with other local conditions made it seem advisable, after careful investiga-

tion, to retain the power plant practically in its old location.

67 *Possibilities of Low Pressure Turbine.* The possibility was considered of increasing the available power output by means of a low-pressure turbine taking steam from old engines but found impracticable because of the extremely advanced age and light construction of the engines. So far as the writer's experience goes this has proved to be the determining factor in a number of cases where this expedient has been considered. The engines in many textile mills are now very old, having served the full period generally allowed to constitute the useful life of such a machine and frequently are of much lighter construction than the best engines of a later date.

68 To install expensive new equipment dependent upon and forming a unit with such engines, does not seem justifiable, however desirable it might be were the engines and flywheel of thoroughly good construction. It was finally decided to abandon the manufacturing floors over the space originally occupied by the old engines, to wall off this portion, do away with one of these engines and to convert this space into the start of a modern steam turbine room.

69 *Special Conditions Requiring Adaptation.* Crossing diagonally under the end wall of the building and extending well under the space which must be occupied by steam turbines were the two tail races of one of the group of waterwheels, and the arches of these old tail races were not of particularly good construction or suitable to form portions of foundations for modern large sized units. Still further, in order to provide for the ultimate capacity of the plant desired by the owners, it was necessary to economize more than ordinarily in the use of room, as space available in this location was much cramped.

70 The solution of the whole problem which at once overcame all these difficulties was found by supporting the main generating units on a system of deep transverse girders whose columns were either carried directly to ledge rock or supported on other girders over the raceway and the supports of which were likewise carried to rock.

71 In this way a basement was secured giving only a small amount less room than that of the floor above and saving for other uses practically all the space usually occupied by foundations, but every inch of which will ultimately be needed for condensers, feed pumps, fan, engines and main piping.

72 The supporting of machines on girders made possible the placing of condensers directly under exhaust nozzles of steam turbines and in the case of the present LeBlanc condenser unit and one future unit, the connecting of condenser discharge through a water sealed pipe directly to tail race, thus considerably decreasing the work which condenser turbine has to do.

73 *The New Equipment:* The first unit of this group which is a General Electric steam turbine of 3244 kw. capacity at 80 per cent power factor, running at 1800 r.p.m. and generating 3-phase 60-cycle current at 600 volts, has now been in service about a year and a half, and has been running constantly in a thoroughly satisfactory manner with no trouble from vibrations or other causes. This is illustrated in Fig. 9 and is specially mentioned because so far as the writer knows this is the first instance where a large machine has been supported by transverse girders. There are several earlier instances where machines of other makes and of large size have been supported by longitudinal girders.

74 The question of securing needed additional boiler capacity has been solved for the present by installing an induced draft fan and economizer, the engine which drives the fan being placed in the basement under the main turbine room floor (Fig. 10) together with the boiler feed pumps, thus grouping all auxiliaries together where they are away from the dust and dirt of the boiler room and under the constant observation of the turbine room attendant.

75 The drips from heating systems in winter and from slashers at all times are returned to a small open heater to which fan engine and condenser turbine exhaust. Water is taken from this reservoir to economizer usually at about 90 to 100 deg. temperature, thus leaving an ample range through which the economizer can and does work.

76 The whole equipment thus forms a compact unit and even with the light loading which at times results from the steam turbine being in part a relay to the waterwheel equipment, the over-all economy of this steam group has been excellent.

77 A connection for stage extraction of steam for use in mill heating is provided on this machine, and the piping connections necessary in the station are already completed, but as the revision of heating system throughout the mills has not as yet been made, this has not been used.

STEAM TURBINE ISOLATED STATION

78 A power plant recently built for the West Point Manufacturing Company, Langdale, Ala., furnishes an example of a complete new development, situated out of the way of and apart from manufacturing buildings and with unlimited space for expansion to meet possible requirements of a series of mill groups. The primary purpose of this development was to furnish a relay to a water power development.

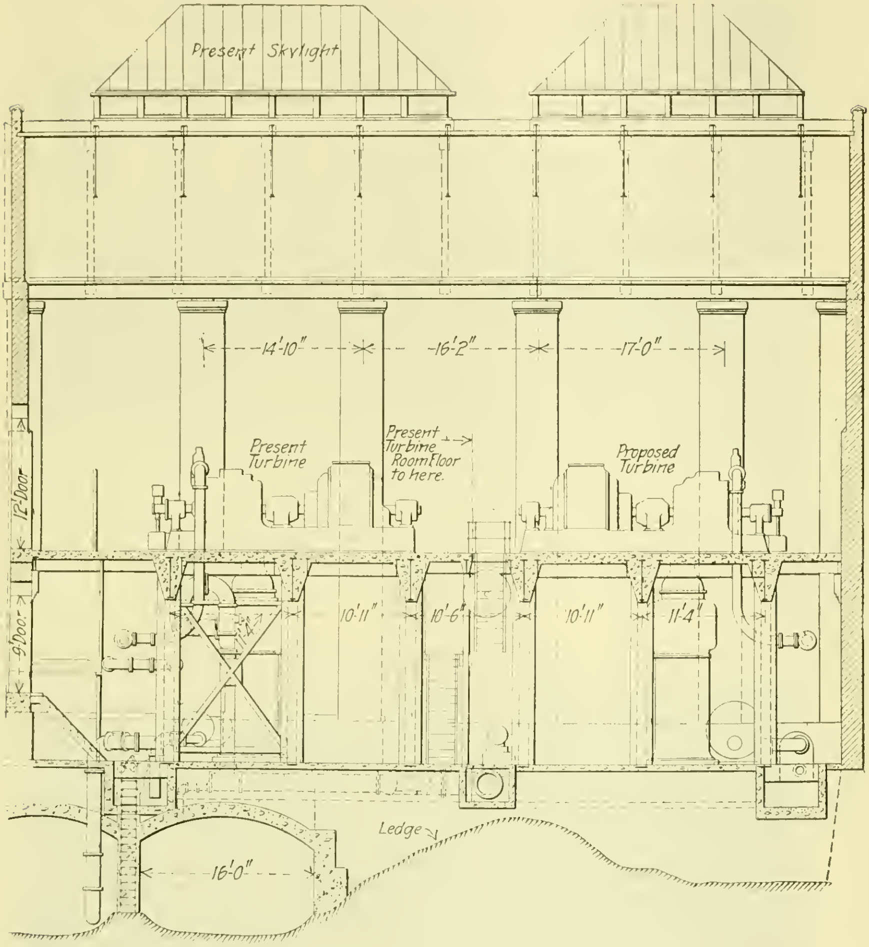

Fig. 9 Sectional Side Elevation of Turbine Room for Boott Mills, showing Support of Turbine by Girders

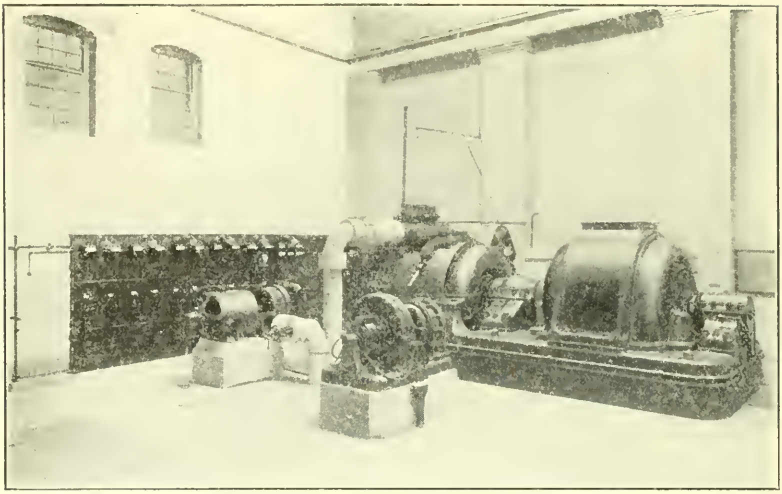

FIG. 10 INTERIOR VIEW OF MAIN FLOOR OF TURBINE ROOM AT BOOTT MILLS

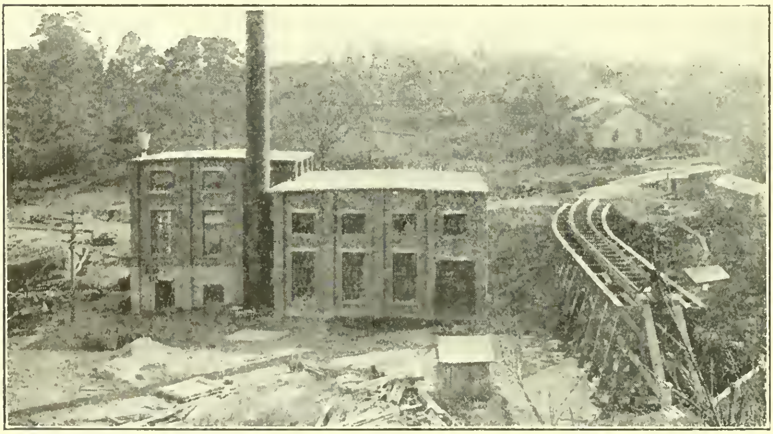

FIG. 11 OUTSIDE VIEW OF TURBINE AND BOILER HOUSE FOR WEST POINT MANUFACTURING COMPANY, LANGDALE, ALA.

79 The old power plant at this group was so situated as to have very limited coal-handling facilities and no room for further extension, while the new location chosen (Fig. 11) is in a natural amphitheater which offers ready means for handling and storing large quantities of coal by merely dumping from a trestle.

80 There is also an unlimited supply of condensing water close at hand. This is taken from discharge of waterwheels and furnishes an element which is lacking or difficult of access at some of the other mill groups. These two combined advantages of location thus make this the natural point for the start of a central station.

81 The unit first installed consisted of two 308-h.p. Heine boilers and a 750-kw. General Electric Company 3600-r.p.m. 2-stage steam turbine connected to a LeBlanc condenser and with generator for 600 volts. As there was good prospect of growth in the immediate future, the boiler house was built large enough for five units and the turbine room (Fig. 12) made sufficient in size for one additional larger unit. The three extra boilers and a 2000-kw. steam turbine are now in process of erection in these spaces, as is also a permanent brick chimney to replace the temporary steel stack which served the first installation.

LARGE PLANT PRODUCING CARPETS AND CARPET YARNS AND HAVING EXTENSIVE DYE HOUSES

82 An installation on which an exceptional number of special problems had to be met was that of the Bigelow Carpet Company in Lowell, Mass., a large corporation engaged in the manufacture of carpet yarns and carpets.

83 This company has extensive dye houses which make a more than ordinarily constant demand for large quantities of low-pressure steam, a portion of which had been furnished by the exhaust from old engines.

84 The irregular grouping of buildings in this plant made any form of mechanical transmission of power particularly difficult and resulted in many complicated drives and heavy friction losses. The plant is situated in the heart of the city where land values are high and a desirable vacant space for a power station was lacking.

85 A portion of its power was being obtained from three

vertical waterwheels, two being in one group, all situated in mill basements and all very old.

86 *Most Advantageous Position for New Station as Schemed by Analysis.* In the power plant analysis which had preceded the undertaking of this development, the many advantages of a combined steam turbine and hydroelectric station located on the space then occupied by the small mill building, in whose basement the two old waterwheels were situated, had been pointed out.

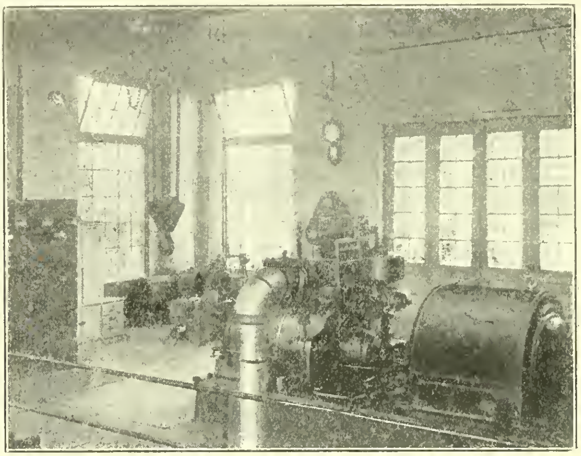

FIG. 12 STEAM TURBINE ROOM, WEST POINT MANUFACTURING COMPANY

87 A system of concrete feeders extending from the main canal to within a few feet of this building had been installed only recently at considerable expense. This building was also located in close proximity to new dye houses then under construction, where large quantities of low-pressure steam would be required. By abandoning the three old waterwheels and making all new development in one location, the old single feeder became an excellent underground tunnel for pipes and electric cables.

88 This tunnel was available in part for a passageway to and a means of getting steam from boiler house to turbine house and throughout as a means both of distributing electric current, and furnished space for a water pipe line from feeder to dye house. The old wheel pit too, became available at small

expense as a much needed central low point to which all plant steam returns could be drained and thence pumped to the boiler house.

89 It was decided to demolish the old building and in its place build one which would accommodate a complete new waterwheel equipment with groups of wheels direct-connected to electric generators, a steam turbine station for present and all anticipated future development and besides provide above the section used for power station, two floors of manufacturing space. This manufacturing space is occupied by picker machinery. The floors match those of adjoining mill building and afford communication to other groups.

90 *Description of New Building and Power Plant Equipment.* One corner of the rectangle otherwise available was taken up by the stair tower of an adjoining mill and in the remaining space switchboard galleries and exciters were located.

91 As shown in Figs. 13-17, the main turbine room floor is situated directly over the waterwheels which are placed in concrete waterways with foundations extending down to bed rock.

92 The waterwheel generators are accommodated in a small space at the far end of the building, and at an elevation corresponding to that of the bottom of the waterways.

93 This portion of work, up to and including the turbine room floor, is of reinforced concrete and was so designed and built that steam turbine units of much greater size than those now required can be placed anywhere in the space as may be desired.

94 The present steam turbine equipment consists of three 750-kw. General Electric units especially designed with top exhausts and capable of being run either condensing in the ordinary way or non-condensing against a back pressure of 6 to 10 lb. These turbines are connected in groups of two with Westinghouse LeBlanc condensers, the space for the fourth turbine unit as yet being vacant.

95 Water for those condensers is taken from the middle waterwheel feeder and after doing its work in condensers, is returned to the feeder where it is later used through waterwheels, so that there is thus no extra use of water for condensing purposes.

96 The steam connections to condensers are controlled by

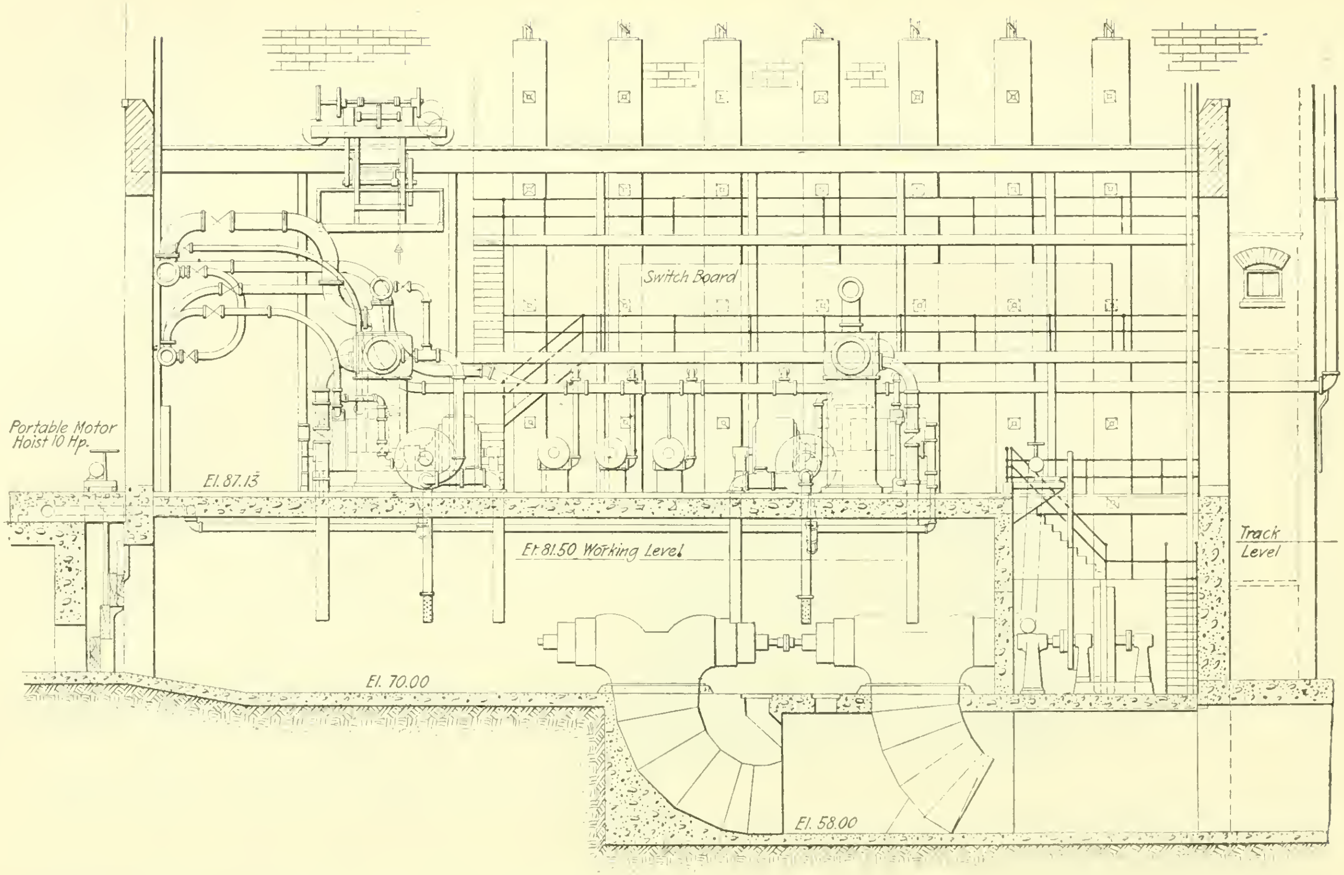

FIG. 13 SECTIONAL SIDE ELEVATION OF COMBINED STEAM TURBINE AND HYDROELECTRIC STATION, BIGELOW CARPET COMPANY

motor-operated gate valves. Besides the connection to condenser, the exhausts of two of these turbines are also connected to a 20-in. low-pressure steam line leading to the dye house and the low-pressure piping system of the plant.

97 All steam-driven auxiliaries in this station were designed to run against a back pressure and are connected to the low-pressure main. Additional connections are provided for the condenser turbines so that they may exhaust direct to the condenser should they have to run for long periods when the dye house is not in operation. The reducing valve connection direct from high-pressure to low-pressure lines is also located in this room.

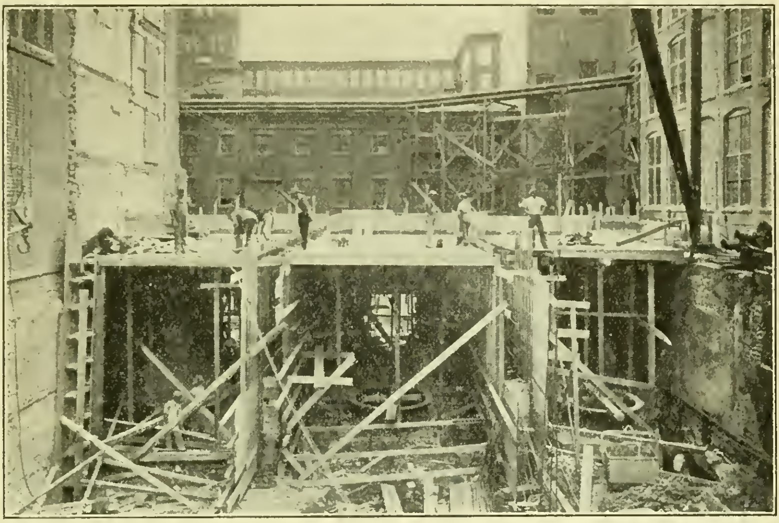

Fig. 14 Waterways, Draft Tubes and Turbine Room Floor, Bigelow Carpet Company

98 The operating switch mechanism for generating and distributing circuits from the steam turbines and three waterwheel generators is all controlled from the lower switchboard gallery, the switchboard carrying the indicating and recording instruments being mounted here. The oil switches are mounted in the gallery above. This portion of the apparatus was furnished by Westinghouse Company.

99 A set of gages from all important lines in the station is

also mounted on a special panel of this switchboard. The attendant thus has the whole working of the station at all times under control and he so adjusts the loads on the non-condensing and condensing machines as to maintain a constant pressure in the low-pressure manufacturing steam line.

100 *Turbine Casing Drainage.* Each of the three steam turbines being of the top exhaust type, is fitted with casing drains and traps to take care of whatever moisture may accumulate in the exhaust casing.

101 It will be interesting to note that the amount of condensation which experience has shown that these traps have to handle when machines are working against a back pressure, or in other words, delivering steam to a low-pressure system, is extremely small.

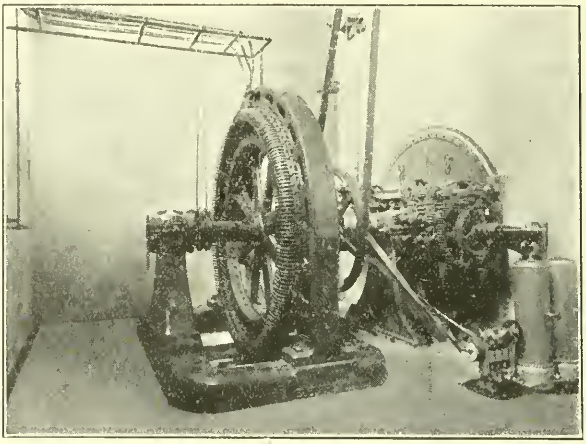

Fig. 15 Waterwheel Generator Pit, showing Generator and Water wheel Gate Mechanism, Bigelow Carpet Company

102 Gage glasses placed on the connection from turbine casing to trap and observed for long periods under running conditions show that there is almost no accumulation of condensation at this point. Whatever moisture may be present when leaving the last row of blades of these turbines practically all goes through with the steam into the low-pressure system.

103 *Moisture in Steam Leaving Turbine.* The writer was much interested to obtain as accurate information as possible

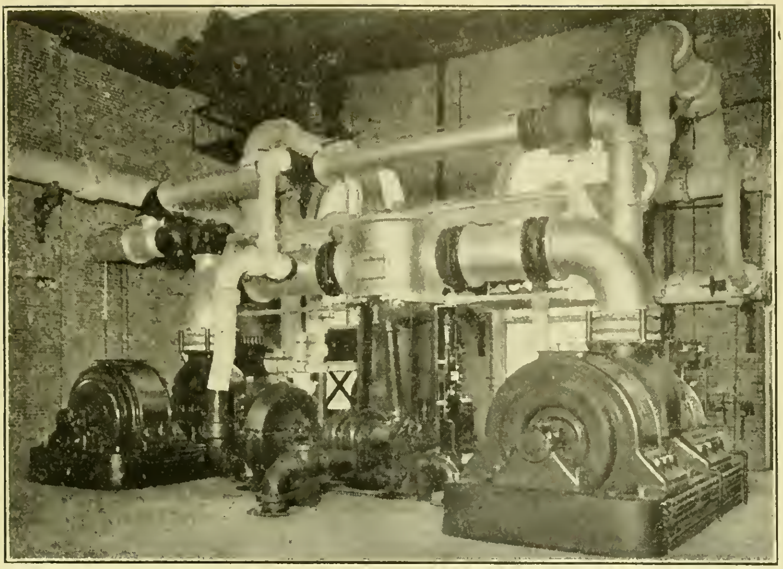

FIG. 16 END VIEW MAIN FLOOR OF TURBINE ROOM, SHOWING CONNECTIONS OF STEAM TURBINES FOR SUPPLYING STEAM USED IN MANUFACTURING, BIGELOW CARPET COMPANY

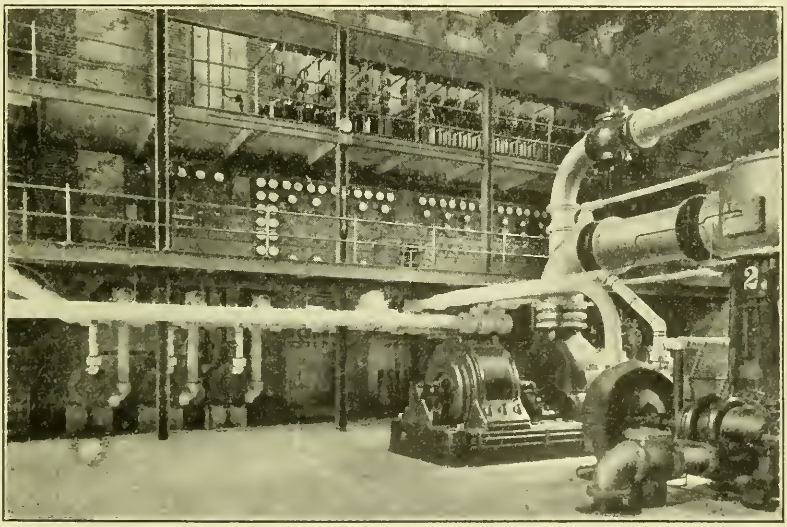

FIG. 17 SIDE VIEW TURBINES AND SWITCHBOARD, BIGELOW CARPET COMPANY

concerning the quantity of this moisture. In order to secure such results a set of tests with one of these turbines running under ordinary mill conditions, exhausting to the low-pressure system, and with loads varying over a wide range, was made January 19, 1912, by Prof. Charles W. Berry, of the Massachusetts Institute of Technology, using for this purpose a constant pressure superheating calorimeter. The steam at throttle of this turbine was slightly superheated, the amount being about 10 deg., as is the usual condition at this plant.

104 The quantities of moisture found were as follows:

With Load in Kw.	Moisture Per Cent
625	4.07
560	4.48
410	4.98
170	5.49

As would be expected, the per cent of moisture increases as the load decreases, the radiation losses of the turbine remaining practically a constant.

105 *Portion of Steam Chargeable to Power.* Adding to this, in the worst case, 5.5 per cent of moisture in steam leaving turbine for manufacturing use, the percentage representing the difference in heat energy between a pound of dry steam at 150 lb. boiler pressure and a pound of dry steam at manufacturing pressure of say, 6 to 7 lb. gage, we have about 9 or 10 per cent. Then increasing this by a liberal allowance for all losses and varying conditions, it seems to the writer that the quantity of steam which should be charged to power purposes when being exhausted from a steam turbine on its way to manufacturing or heating as above, does not exceed 12 per cent and that for the ordinary pressure ranges with practically dry saturated steam from boiler this value may well be used.

106 This station has been in full operation in the manner indicated and furnishing all power to the plant since June 1911. The entire substituting of motor for mechanical drives has in this case proved very satisfactory, as the grouping of buildings was such as to present many difficulties with any other system.

107 The quantity of work being done in the dye house sections is constantly increasing, and the portion of total power done on non-condensing turbines has proportionately increased.

ELECTRIC TRANSMISSION AND RELAY POWER

108 Lest it should be thought that the writer unduly favors electric transmission, inasmuch as most of the examples above mentioned have to do with the use of this method, it should be mentioned that this is not the case. The examples given were all taken from the work of his office during a short period which happens to cover a group of cases where flexibility was one of the most important determining factors. In these cases flexibility was most advantageously secured through electric transmission.

109 The greater portion of the plants, however, still contain older groups on direct mechanical drives. Where such drives are now in position, are reasonably direct and well constructed and where loads are fairly constant, their abandoning would certainly in many cases be undesirable.

110 Any new installation has to be considered with reference to its own special group of conditions and requirements taken together as a whole and in the writer's opinion no general rule can be laid down, for almost certainly there will be found cases where each form of transmission will be the cheapest and best.

RELAY POWER

111 By relay power is meant the power to supplement the usual or cheapest form of power and to take its place in case of failure. So far as the demands of textile mills are concerned, such power may be subdivided into two groups because of the different conditions affecting them.

112 Relaying almost always serves to relieve (*a*) either a shortage of water power which can be definitely foreseen and will take place in all plants in a given vicinity at the same time or (*b*) to furnish an added source of power available during times of accident to the ordinary equipment. It is then merely a form of insurance against events which can be expected to take place at no regular interval or predetermined time.

113 Mills using water power have usually had to provide for the first-mentioned kind and a group of such mills on a given stream must each provide for about the same proportion of such power.

114 *Auxiliary Equipment as Insurance Against Shutdowns.* When we come to the second form, however, it has in times

past been rather usual for textile plants to make little provision for added units to meet conditions of accident. This is due to the long period of shutdown during each 24 hours, allowing for ordinary repairs, and to the greatly increased cost for maintaining such equipment.

115 There is, however, a growing tendency to make some provision for added sources to relieve the danger of failure by the ordinary power. In many cases managers would be glad to increase this form of insurance greatly, could a proper solution of the question of cost be found.

116 In order to protect fully a group of independent plants would require duplicate equipment for their whole combined output, but this same group, if united together and provided with one source of relay power, could be, practically speaking, fully protected by an equipment very much less than that of the combined plants. The proportionate quantity of such equipment required to give protection would decrease rapidly with an increased number of plants served.

117 This supplying at low cost power connections, serving merely as an insurance against accident to the ordinary equipment, is a phase of power plant development which it seems to the writer should receive more attention than it has in the past.

118 *The Public Service Station as a Means of Supplying Insurance.* The public service corporation would seem to be in a position to undertake this service at rates which would be attractive and profitable both to itself and to those it would serve. With few exceptions, however, so far as the writer knows, little or nothing is being done by them at present time to develop this form of business. For the most part they at present insist upon a minimum charge which is practically the equivalent of all fixed charges on an equipment great enough to handle the full connected load of each unit served.

119 Such rates are of course in no way attractive to the manufacturer as under these conditions he can as well supply the added equipment in his own station and have it where he can control its use.

120 *Private Stations Serving Groups of Owners.* In certain sections of the country there is a growing recognition of the advantages of properly located groups of manufacturers combin-

ing to produce their power in a single station of large size rather than in individual stations.

121 There seems no reason why this idea should not prove of advantage in many cases with textile manufacturers. Such stations could be jointly maintained and run by them with very little increased overhead or supervision charges as there would be no expensive sales, inspection and advertising departments. The securing of relay power in this manner as an insurance against expensive shutdowns should be particularly attractive.

122 There are of course many difficulties in the way of carrying out such a scheme, principal among them that of having a group of individuals ready to act at a given time and the securing of rights of way for transmission lines. In connection with the latter strenuous opposition would sometimes have to be anticipated. The writer does not himself expect to see rapid developments along these lines, but nevertheless, believes that there will and should be a growing recognition of the wastefulness of some of our present seemingly necessary expenditures for duplicate power developments.

123 It seems reasonable that much more than in the past such portions of power required by textile mills as cannot be made a by-product of some other process will be produced in large central stations with the coal-burning units well removed from centers of population and so located as to give access to coal cheaply handled.

124 As electric energy can now be transmitted for long distances with comparatively small loss, there seems no reason why such stations for serving inland cities or groups of cities, like Lowell and Lawrence, should not be actually located at tidewater, and receive the full advantage of the reduced coal transportation rates which would thus result.

CASE CARBONIZING

BY MARCUS T. LOTHROP

ABSTRACT OF PAPER

The case-carbonizing process requires an increasing of the carbon content of the surface of the material treated while the original composition of the interior is retained. It is essential that the details of the operation be understood to secure uniformity of product, and these researches were undertaken with the object of determining the causes of irregularity and the precaution which must be employed to secure uniform results.

The experiments undertaken were to determine:

a The effect of heat treatment in case-carbonizing steels upon the physical properties of the metal.
b The effect of temperature used in case carbonizing.
c The effect of heat treatment after case carbonizing.
d The effect on the rate and depth of carbonization of the composition of the steel being case carbonized.
e The efficiency of case-carbonizing materials
f The rapidity with which heat is conducted through carbonizing materials.

The conclusions deduced in each case are listed in the body of the paper under the headings corresponding to the several experiments, and the detailed results of the experiments are tabulated for further reference in the appendices.

CASE CARBONIZING

By Marcus T. Lothrop, Canton, Ohio

Junior Member of the Society

During the last decade the better understanding of the constitution of ferrous alloys and the improvement in apparatus have led to great advances in the commercial heat treatment of steel. The progressive manufacturer today carefully inspects the steels as received and before he allows them to enter his shop knows their composition and their physical condition. After the raw material is fabricated into finished parts, these parts are heat treated to show the desired physical properties. This operation of heat treatment must be conducted with certainty and uniformity and an inspection of existing conditions indicates that the uniformity is much greater when the manufacturer does not change the composition of the metal being heat treated. Case carbonizing, however, requires that the carbon content of the material on the surface be increased while the original composition of the material in the interior is retained. The operation requires that the user, in a sense, make his own steel and the details of the operation must be understood to the end that the operation may be conducted with the minimum cost and with the maximum certainty and that variations and irregularities shall not creep into the product.

2 The investigations indicate that, commercially, the case-carbonizing operation can be performed in such a manner that the final result will be absolutely uniform and that the advantages which accrue from case carbonizing represent the most efficient manner of obtaining the desired physical properties in many constructions. It was with the object of pointing out the causes of and the results effected by irregularity and the means

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The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

and precautions which must be employed to produce uniformity, that these researches were undertaken.

3 The following points were studied:

- *a* The effect of the composition of the steel on the rate and depth of carbonization.
- *b* The effect of heat treatment on case-carbonized steel.
- *c* The effect of time on the rate and depth of carbonization.
- *d* The effect of temperature and of composition of the steel on the depth and nature of carbonization.
- *e* The efficiency of various commercial carbonizing materials.
- *f* The effect of depth of case carbonizing on the physical properties of the finished materials.
- *g* The commercial thermal conductivity of the various case-carbonizing materials.
- *h* The efficiency of copper plating in preventing carbonization.

The materials used are summarized in Table 1.

EXPERIMENT NO. 1

THE EFFECT OF HEAT TREATMENT IN CASE-CARBONIZING STEELS (TABLES 12-21)

4 The determination of a test method which would indicate the physical properties of the metal tested required considerable trial. A case-carbonized piece of metal, consisting of a hard exterior and a soft and tough interior, is so dual in its physical characteristics that tensile tests are not adequate. This is due to the fact that when the case which is hard and brittle fails, the pull is suddenly applied to the reduced section of the core and causes it to fail so quickly that the expected ductility of the material fails to appear.

5 The test finally decided upon was a transverse test which has the advantage of approaching the condition of stress generally encountered in service. The test piece adopted was ½ in. round, 12 in. long, with 10 in. between supports, loaded at the center. A round section gives a case-carbonized zone of uniform depth in all parts and for this reason is superior to the flat section which has excessive depth in the corners. The 10-in. center to center distance between supports allows loads large

enough to be measured accurately and gives a deflection at rupture great enough to be easily measured. By means of this test, pieces of the same material and heat treated in identically the same way give results for strength and toughness which check closer than by any other method tried.

HEAT TREATMENT METHOD

6 These test pieces were packed in wrought-iron pots 23 in. long by 5 in. round, with 0.25 in. walls, so that each was 1 in. from the inside wall of the pot and 1 in. from each other. Case-carbonizing compound No. 1[1] was used in this experiment. The precaution of having everything clean and dry and the pots well and uniformly tamped down was carefully observed, and identical conditions of rate of heating, cooling and temperature maintained. The temperature was 1700 deg. fahr. with a variation of not greater than plus or minus 15. The test pieces after case carbonizing were cleaned and reheated in a neutral atmosphere to the temperatures given in Tables 12-21, Appendix No. 1, which are correct to plus or minus 10 deg. fahr. Table 2 sum-

TABLE 1 MATERIALS USED

No.		C	Si	Mn	P	S	Cr	Ni	V
2	0.15% C basic open hearth......	0.15	0.02	0.68	0.054	0.043	....	....	....
3	0.20% C basic open hearth......	0.20	0.02	0.66	0.038	0.044	....	....	....
4	0.25% C basic open hearth......	0.25	0.04	0.61	0.038	0.046	....	....	....
5	0.10% C Halcomb electric furnace nickel steel..................	0.10	0.13	0.66	0.010	0.025	....	3.60	....
6	0.15% C Halcomb electric furnace nickel steel..................	0.15	0.09	0.70	0.009	0.013	....	3.54	....
7	0.20% C Halcomb electric furnace nickel steel..................	0.20	0.18	0.69	0.007	0.013	....	3.59	....
8	0.25% C Halcomb electric furnace nickel steel..................	0.25	0.21	0.74	0.009	0.009	....	3.55	....
9	0.31% C Halcomb electric furnace nickel steel..................	0.31	0.18	0.70	0.009	0.010	....	3.70	....
10	0.20% C Halcomb electric furnace chrome vanadium steel........	0.20	0.13	0.50	0.009	0.010	1.00	....	0.15
11	0.50% C Halcomb electric furnace chrome vanadium steel........	0.49	0.15	0.80	0.009	0.021	1.18	....	0.19
12	0.50% C Halcomb electric furnace chrome nickel steel...........	0.48	0.16	0.44	0.009	0.009	0.98	2.02	....

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[1] The analyses of these and other compounds are given in Table 7.

TABLE 2 INCREASES DUE TO HEAT TREATMENT

No.	Increase in Strength with Double Heat Treatment, Per Cent	Increase in Toughness with Double Heat Treatment, Per Cent	Drawing Single Treatment Increase		Drawing Double Treatment Increase		Annealing Single Treatment Increase		Annealing Double Treatment Decrease	
			Strength, Per Cent	Toughness, Per Cent	Strength, Per Cent	Toughness, Per Cent	Strength, Per Cent	Toughness, Per Cent	Strength, Per Cent	Toughness, Per Cent
2	23.0	105.0	12.0	25.0	....	....	10.0	26.0	....	....
3	25.0	56.0	21.0	4.0	....	....	16.0	4.0	....	....
4	12.0	80.0	12.9	18.2	....	....	14.3	18.4	....	....
5	20.0	10.8	24.8	10.8	9.1	20.0	2.4	6.5	....	....
6	6.1	....	23.0	17.8	11.0	10.0	4.1	....	....	....
7	5.6	13.6	27.4	20.5	14.5	16.0	4.0	....	1.5	6.0
8	6.9	19.0	29.6	43.0	17.0	23.8	1.5	2.4	0.7	8.0
9	6.9	....	34.4	27.6	25.0	40.5	4.1	2.1	3.3	....
10	10.7	92.5	14.3	25.0	16.7	13.6	2.8	7.0	....	....
Average carbon steel....	20.0	80.0	15.0	16.0	....	....	13.0	16.0	....	....
Average nickel steel........	8.0	14.0	27.6	24.0	15.3	20.0	3.6	2.0	1.1	3.0

marizes the results due to reheating and annealing, showing the increases in strength and toughness. Table 3 gives the temperatures at which the case ceases to be file hard as a result of drawing. All the tests recorded in the tables are the average of closely agreeing duplicates, and were made on test pieces which after heat treatment were file hard.

GENERAL CONCLUSIONS, EXPERIMENT NO. 1

7 A study of the tables leads to the following conclusions:

a Strength and toughness decrease with increasing depth of case. This is important commercially for with a certain steel and a certain heat treatment uniformity of strength and toughness is impossible if the depth of case varies.

TABLE 3 EFFECT OF DRAWING IN CASE-CARBONIZED STEELS

No.	Steel No. Grade	Temperature at which Case Ceases to be Glass hard, deg. fahr.
1	1—.10 per cent—C—Bessemer	425
2	2—.15 per cent—C—O. H.	425
3	3—.20 per cent—C—O. H.	425
4	4—.25 per cent—C—O. H.	425
5	5—.10 per cent—C—E. F. Ni.	425
6	6—.15 per cent—C—E. F. Ni.	425
7	7—.20 per cent—C—E. F. Ni.	425
8	8—.25 per cent—C—E. F. Ni.	425
9	9—.30 per cent—C—E. F. Ni.	425
10	10—.20 per cent—C—E. F. Cr. V.	450

The temperatures are such that the case can just be touched with a new fine file.

b The critical depth of case at which maximum brittleness and minimum strength occur, depends upon the initial carbon content of the steel which is case carbonized. The higher the original carbon content, the smaller the ratio of depth of case to diameter of core at the point where this brittleness occurs. The commercial importance of this point is that the manufacturer using the steel for case carbonizing should know its composition and should vary the depth of the carbonizing effect as demanded by the composition of the metal being used.

c Strength and toughness increase with double heat treatment. This result is to be expected, for the dual nature of case-carbonized steel requires two thermal treatments properly to refine the composite metal.

d Strength and toughness increase when the temper is drawn at 380 deg. fahr. This operation does not decrease the hardness but relieves in a greater or less degree the strains set up in hardening and makes for uniformity as well as more pronounced physical properties.

e Annealing after case carbonizing is a doubtful operation. The physical improvement with single heat treatment may be obtained by resorting to double heat treatment, for the results will be better and the time consumed in the operation much shorter. Annealing does not improve the double heat treated pieces and may even cause loss of strength and toughness, due to the decarbonization which may take place. Annealing is not to be recommended.

f The case-carbonized steel will, with either double or single heat treatment, become file hard at temperatures too low for the best development of the maximum strength and toughness. Although seemingly paradoxical, the dual nature of the steel after case carbonizing will explain this established fact, when one recalls that the transformations of the exterior are first completed at low temperatures, and that those of the interior are last completed at higher temperatures.

g All steels lose their file hardness when drawn at 425 deg. fahr., save chrome vanadium which loses it at 450 deg. fahr.

h The ideal heat treatment for a case-carbonized steel may be assumed to be: Case carbonize to the thinnest possible depth of case demanded by the conditions of service; reheat for the core; quench in a suitable fluid; reheat for the case; quench in a suitable fluid; draw the temper as far as the conditions of service will permit.

i Case-carbonized steel parts, such as gears, etc., may be stronger, tougher and harder than similar parts made from oil-hardened steels.

EXPERIMENT NO. 2

MICROSCOPIC EXPERIMENT ON THE EFFECT OF HEAT TREATMENT AFTER CASE CARBONIZING

8 The object of this experiment was to determine, if possible, the constitutional difference between a case-carbonized bar of steel which had received a single heat treatment (reheating being a compromise, in that the temperature selected may be a little too high for the best refinement of the carbonized zone, and a little too low for the best refinement of the core), and a piece of the same metal carbonized in the same manner which had received a double heat treatment, the first reheating designed to be the best possible for the interior or the core and the second reheating designed to be the best possible for the case or carbonized zone.

9 Steels Nos. 3, 7 and 10 were case carbonized at 1300, 1400, 1500, 1600 and 1800 deg. fahr., and were treated with the best single and double heat treatments as detailed in Experiment No. 1. The specimens were etched for 10 seconds in a 1 per cent nitric acid solution. All specimens were martensitic with double heat treatment.

a The martensite was easily distinguished in steel No. 3 (0.20 per cent carbon open hearth) at 100 diameters magnification.

b The martensite was distinguished with difficulty in the No. 7 steel (0.20 per cent carbon electric furnace nickel steel) at 100 diameters magnification.

c The martensite was not distinguishable in steel No. 10 (0.20 per cent carbon chrome vanadium steel) at 100 diameters magnification. Martensite was barely distinguishable in this steel at 400 diameters magnification.

10 With single heat treatment

a The martensite was very easily distinguished in the No. 3 steel at 100 diameters magnification.

b The martensite was easily distinguished in the No. 7 steel at 100 diameters magnification.

c The martensite was just barely distinguishable in the No. 10 steel at 100 diameters magnification. At 400 diameters the martensite with this steel was easily distinguished.

11 The results of this experiment indicate that the double heat treatment increases the strength and toughness of case-carbonized steels due to the smaller and finer martensitic structure developed.

EXPERIMENT NO. 3

THE EFFECT OF TEMPERATURE USED IN CASE CARBONIZING

12 The object of this experiment was to determine the lowest temperature at which case carbonizing begins; the lowest temperature with each of the three different types of case-carbonizing steels (straight carbon and alloy, all of the same carbon content) at which it might be most efficiently conducted in practice; and the best temperature for the case carbonizing of steel.

13 The following steels were used:

a No. 3 0.20 per cent carbon open-hearth carbon steel
b No. 7 0.20 per cent carbon electric nickel steel
c No. 10 0.20 per cent carbon electric chrome vanadium steel

14 The test pieces were ½ in. round and 6 in. long, properly numbered for identification; three, one of each grade of steel, were packed in a pot for each run. The pot used in this experiment was made of 4 in. round wrought iron 8 in. long with ¼ in. wall. A solid steel plug was welded into one end, and the other end was fitted with a removable sheet-steel cover which was drilled in the middle with a hole large enough to accommodate the electro-thermo-couple protection tube. The test pieces were placed in the pot, equally spaced about the electro-quartz protection tube in the center and packed tightly in case-carbonizing compound No. 1. All heating was done in a gas-fired muffle furnace. Temperature readings of the furnace temperature and the inside temperature of the pots were taken every 15 minutes. Standardized platinum platinum-rhodium thermo-couples were used. Six runs of 5 hours each at a definite temperature were made, using the same apparatus and procedure. The six runs were at the following temperatures in deg. fahr.

No. 1	1300	No. 3	1500	No. 5	1700
No. 2	1400	No. 4	1600	No. 6	1800

15 The furnace was started cold in each case and time was counted from the time the pot couple showed a temperature in-

side the pot corresponding to the temperature for which the run was intended. After this, the pot temperature was maintained as close as possible to this point for an uninterrupted period of 5 hours, at the end of which time the pot couple was removed, the pots pulled from the furnace and allowed to become cold.

16 The temperature records of this series of experiments are given in Tables 22-27 of Appendix No. 2. Following these tables in the same appendix is a detailed analysis of the microscopic examinations of the various test pieces after being subjected to the heat treatment and specimen microphotographs are reproduced in Figs. 18 to 30, shown full size.

17 The conclusions to be drawn from this analysis follow herewith together with a tabulated statement (Table 4) of the thickness, in millimeters, of the zone of excess cementite, of the eutectic, of the gradation zone and the total penetration of carbon in millimeters for the different steels.

CONCLUSIONS FROM MICROSCOPIC INVESTIGATION OF OPEN-HEARTH STEELS CASE CARBONIZED AT 1300 TO 1800 DEG. FAHR.

18 In the specimen treated at 1300 deg. fahr. a small zone of carbonization was clearly separated from the interior material having a great number of penetrating bands containing the so-called nitrides in very minute needles. In the gradation zone these nitrides appeared in large quantities in larger needles.

19 In the specimen treated at 1400 deg. fahr the carbonized zone was slightly thicker and the gradation zone larger, giving a more gradual division between the carbonized and interior material. The penetrating bands were present as before in large numbers. Nitrides were still present inside the eutectic zone, but not in such great numbers as in the first specimen.

20 In the specimen treated at 1500 deg. fahr. the carbon penetration was slightly deeper, and a zone of excess cementite was found on the outside, which appeared as large irregular areas and as polygonal outlines. No nitride needles of any kind were visible.

21 The specimens treated at 1600 deg. fahr. and 1700 deg. fahr. showed still deeper penetration of carbon, greater excess of cementite, and a larger number of cementite needles.

22 At 1800 deg. fahr. there was a much greater penetration of carbon with an outer zone of excess cementite, but this cement-

ite did not occur in polygonal outlines, except in a few places. The general appearance of the carbonized zone was that of a eutectic, containing areas of excess cementite, grading to the normal carbon content of the interior.

CONCLUSIONS FROM MICROSCOPIC INVESTIGATIONS OF NICKEL STEELS CASE CARBONIZED AT 1300 TO 1800 DEG. FAHR.

23 In this series there was a slight carbonization in the specimen treated at 1300 deg. fahr. only, and no penetration bands similar to those of the open-hearth series which contained nitrides.

TABLE 4 SUMMARY CASE-CARBONIZING EXPERIMENT NO. 3

Temperature. Deg. Fahr.	Excess Cementite			Thickness of Eutectic			Gradation Zone			Total Penetration of Carbon		
	O. H.	Ni	Cr Va	O. H.	Ni	Cr Va	O. H.	Ni	Cr Va	O. H.	Ni	Cr Va
1100	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
1200	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
1300	0.008	.....	trace	0.090	.....	.....	0.034	0.180	0.016	0.140	0.030	0.180
1400	0.090	.....	0.009	0.120	0.203	0.030	0.188	0.390	0.300	0.410	0.590	0.420
1500	0.128	0.056	0.285	0.150	0.278	0.135	0.420	0.510	0.338	0.710	0.830	0.770
1600	0.135	0.158	0.375	0.210	0.390	0.180	0.450	0.683	0.278	0.770	0.990	0.890
1700	0.225	0.293	0.465	0.338	0.488	0.248	0.600	0.975	0.450	1.160	1.560	1.370
1800	0 293	0.390	0 338	0.450	0.390	0.405	0.683	0.900	0.675	1.640	2.030	1.750

The measurements in this table give the width of zones in millimeters.

24 In the specimen treated at 1400 deg. fahr. there was a better penetration of carbon, but no higher carbon content on the outside than eutectic. Several of the penetrating bands were present.

25 At 1500 deg. fahr. there was a still greater penetration of carbon, but little if any excess cementite on the outside, and few of the penetration bands were present in this specimen.

26 In the 1600 deg. fahr. specimen there was a normally increasing depth of case, with a slight amount of excess cementite on the outside and bands of penetrating material.

27 In the specimen treated at 1700 deg. fahr. there was still greater penetration of carbon and an increase in the excess cementite on the outside of the carbonized zone. In this specimen there were also a few penetrating bands. The cementite existed as crystal boundaries and as needles in the crystal cleavages.

28 In the specimen treated at 1800 deg. fahr. a great deal of excess cementite was found on the outside, as well as greater penetration of carbon and no penetrating bands. This series differed from the open-hearth series in that the excess cementite and the excess ferrite in their respective zones did not tend so uniformly to form the polygonal outlines as they did in the open-hearth series. Also either no nitrides were present in this series, or, if they were in the penetrating bands, they were in a different form from any nitrides so far observed.

CONCLUSIONS FROM MICROSCOPIC INVESTIGATION OF CHROME VANADIUM STEELS CASE CARBONIZED AT 1300 TO 1800 DEG. FAHR.

29 In this series, the specimen treated at 1300 deg. fahr. showed a small zone of carbon, the highest carbon content on the outside being about eutectic.

30 In the specimen treated at 1400 deg. fahr. there was a very slight rim of cementite and pearlite in about equal amounts, while at 1500 deg. fahr. the excess cementite was found in two distinct zones, the outer zone containing the cementite in large globules intermingled with pearlite, the inner zone of polygonal outlines of cementite surrounding pearlite areas.

31 At 1600 deg. fahr. the cementite appeared in two zones as at 1500 deg. fahr.

32 At 1700 deg. fahr. the double zone appearance of the cementite was not so marked, and at 1800 deg. fahr. only one zone contained the cementite in polygonal outlines. The polygons as a rule appeared to be larger towards the inner part of the zone.

GENERAL SUMMARY OF THE EFFECT OF TEMPERATURE IN CASE CARBONIZING

33 The following is a general summary of the effect of temperature in case carbonizing:

a The depth of case carbonizing effect in a given time increases with the temperature.

b The carbon content in the case-carbonized zone in a given time increases with the temperature. This affords the commercial opportunity of varying the depth of case-carbonizing effect as desired by changing the time, and of varying the carbon content of the case-carbonized zone by changing the case-carbonizing temperature. It is poor practice to raise the temperature of the case-carbonizing operation for the purpose of reducing the time which the operation consumes, for this will increase the maximum carbon content of the case-carbonized zone and may change the character of the physical properties of the finished product.

c The minimum temperature at which uniform penetration of carbon can be obtained with case carbonizing seems to be 1500 deg. fahr. With lower temperatures the carbon maximum is not uniform in its distribution and the penetration too slow.

d The presence of excess cementite in the form of needles within the pearlite grains, so conspicuous in nickel and carbon steels, indicates a temperature during case carbonizing which overheats the metal being so treated.

e Nickel steel gives the greatest total penetration and greatest gradation zones.

f Chrome vanadium gives the highest carbon maximum and finest grained steel.

g When case carbonizing is conducted at 1400 deg. fahr. or lower, the carbon maximum does not exceed the eutectic. As the case-carbonizing temperature increases, a zone of hypereutectic composition develops which increases in depth with the temperature. The depth of this zone becomes important when sufficient warpage takes place in the piece being hardened to demand grinding after the final heat treatment. It has been demonstrated often that a carbon content of more than 1 per cent in the surface of a case-carbonized part is undesirable when there is the least shock to be reckoned with, because in such instances the excess of carbon exists in the form of films of carbide between the grains of the metal which are brittle and hard, weakening the steel and rendering it liable to fail by flaking off or spawling. Therefore it would be inadvisable to select a practice or a material which produces such a condition in the surface. On the other hand if the carbon percentage is lower than 0.90 to 1.00 per cent the case will not show its maximum hardness. Therefore, when pieces are to remain unground practice should aim at a production of a 0.90 per cent carbon surface but when a portion of the steel is to be ground off, after the heat treatment, it is advisable to drive in an excess of carbon so as to leave this optimum percentage of 0.9 to 1.00 per cent in the final surface.

h All carbonized zones widen with increase of temperature. While wide gradation zones are desirable from the point of view of tenacity of the case to the core, other considerations enter into the commercial side of the problem. The first is the cost of maintaining the high heats. The factors of fuel cost and increased furnace up-keep cost at the upper temperatures are self-evident. The second is the difference in the nature of the case. Inspection will show that at 1500 deg. fahr. the width of the eutectic zone is considerably greater than that of the hypereutectic zone and the hypoeutectic zone of the decreasing carbon content is wider still, which means, of course, that the case carbonizing is proceeding gradually and that the case and core are merging into each other by gentle degrees. At 1600 deg. fahr. the ratio of the high carbon exterior zone to the inner zones rises and at 1700 and 1800 deg. fahr. the increase is more marked still.

The meaning of all this is that as the temperature goes up carbon is forced by the heat, as it were, to rush into the surface of the steel faster than it can be assimilated by the adjacent grains and so the outside layers at the high heats are gorged with carbon which has not time to diffuse gradually toward the center of the case. This condition of affairs is highly undesirable, because the product then shows a sharp demarcation between case and core, the transition from one material to another being too abrupt; the properties necessarily vary accordingly and in the finished piece have a tendency to set up high physical strains. This causes flaking off of the case from the core, a phenomenon familiar enough to those engaged in the trade. The remedy is to case carbonize at a moderate heat for a length of time sufficient to obtain a reasonable depth of case with gradual transition from case to core. This temperature appears in this experiment to be about 1650 deg. fahr. with open-hearth carbon steel, 1600 deg. fahr. with electric nickel steel, and 1650 deg. fahr. with electric chrome vanadium steel.

In deciding upon the best carbonizing temperature, the effect of heat on the size of grain in the steel must also be considered. This as is well-known, is a very decided reaction, the grain coarsening rapidly with increasing temperatures. In this connection a careful investigation showed marked contrast between the behavior of ordinary carbon steel and the two alloy steels; the former began to coarsen rapidly at 1500 deg. fahr.; nickel steel also showed the same effect at the same temperature but in a much less marked degree and chrome vanadium steel retained its fine grain up to 1700 deg. fahr. and over, not reaching even at 1800 deg. fahr., the grain size of other steels. This is in line with the common belief that chromium and vanadium retain fineness of grain in steel.

EXPERIMENT NO. 4

THE EFFECT OF THE COMPOSITION OF THE STEEL BEING CASE CARBONIZED ON THE RATE AND DEPTH OF CARBONIZATION

34 The object of this experiment was to determine by chemical analysis the depth of carbonizing effect and carbon content of several zones of case carbonizing effect as affected by the composition of the steel being case carbonized.

35 Test bars 8 in. long and 1½ in. round were prepared by machining on centers in a lathe and packed in case-carbonizing compound No. 1, care being taken to see that all bars were evenly spaced. The bars were case carbonized at 1700 deg. fahr. for 7 hours. The pot was then removed from the furnace and allowed to become cold, when it was unpacked and the bars removed and cleaned. The bars were set up in a lathe and cuts 0.005 in. deep taken. The sample thus obtained was analyzed by combustion and in duplicate for carbon, and the results will be found in Table 5. The information obtained from this experiment indicates that:

TABLE 5 EFFECT OF COMPOSITION OF STEEL IN CASE COMPOSITION AFTER CARBONIZING

No.	Material	Percentage of Carbon in Cuts of Various Depths, In.					
		0 to 0.005	0.005 to 0.010	0.010 to 0.015	0.015 to 0.020	0.020 to 0.025	0.025 to 0.030
1	No. 1 0.10% C Bessemer	0.90	0.63	0.32	0.16	0.13	0.11
2	No. 3 0.20% C O. H.	0.95	0.78	0.47	0.27	0.20	0.20
3	No. 6 0.15% C E. F. Ni	0.90	0.65	0.44	0.28	0.19	0.15
4	No. 9 0.30% C E. F. Ni	0.95	0.75	0.55	0.34	0.30	0.30
5	No. 10 0.20% C Cr Va	1.22	0.78	0.49	0.30	0.20	0.20

a The higher the carbon content of the steel case carbonized, the higher will be the maximum carbon content of case when case carbonizing is performed at a given temperature for a given time.

b The original carbon content does not affect the depth of case under the conditions of this experiment.

c The nickel and carbon steels behave similarly in regard to carbon maximum.

d Chrome vanadium steel takes a higher carbon maximum than nickel steel and carbon steel under the same case-carbonizing conditions.

EXPERIMENT NO. 5

EFFICIENCY OF CASE-CARBONIZING MATERIALS

36 This experiment was undertaken to determine the commercial efficiency of case-carbonizing materials and to show the elements which affect their value commercially.

37 Fourteen case-carbonizing materials were selected, representing the most prominent compounds used in this country. Fifty-pound samples were obtained on the open market from the several manufacturers, with prices, general description, etc. As soon as received average samples of about 1 pint each were taken, put in stoppered bottles, and sent to the chemical laboratory for analysis and specific gravity determination. These compounds are designated as Nos. 1, 2, 3, etc., in Table 4. The bars used in making up the test pieces (Table 1) were selected by chemical analysis and were all of 0.20 per cent carbon.

38 The test pieces were of three different types:

a Solid cylinders 6 in. long by 1.344 in. in diameter. These were cut to length and turned from 1⅜ in. diameter hot rolled and annealed bars; set up on centers in lathe and rough turned true and straight and close to finish size; finished absolutely straight and to size by grinding on centers. This left the bars entirely bright and with an almost polished surface ready for carbonizing.

b Hollow cylinders 6 in. by 1¼ in. outside diameter by ¼ in. thick wall. These were also made from 1⅜ in. diameter, hot rolled and annealed bars, turned to size and bored; not ground.

c Solid rod ½ in. in diameter by 6 in. long. Samples were cut to size from ½ in. in diameter (cold drawn) bars and sand-blasted, and all test pieces properly numbered for identification. Carbonizing pots used were new and of cast iron, rectangular in shape, inside dimensions 10 in. by 6¼ in. deep, with ½ in. thick wall. They were supported on legs ¾ in. off furnace floor and prevented from touching each other in furnace by a boss 1½ in. long on each end (Fig. 1).

39 Packing the pots was done as carefully and as nearly uniform in each case as possible. Three test pieces, one of each type, were used in each pot and were spaced as indicated.

40 A layer of carbonizer about 1½ in. thick was put in first and tamped down, then the lowest bar with more carbonizer, etc., tamping in snugly throughout, and so on until the pot was full. The cast-iron covers were luted on with clay, small "tell-tales" inserted in one end, and pots put into the furnace.

TABLE 6 PHYSICAL AND PRICE DATA ON CARBONIZING MATERIALS

Carbonizing Material No.	Weight of 1 Pt., Grains	Approximate Specific Gravity	Weight of 1 Cu. Ft., Lb.	Volume of 1 Ton, Cu. Ft.	Cost per Cu. Ft.
1	450	0.954	59.4	33.7	$1.84
2	365	0.774	48.1	41.5	1.30
3	438	0.930	57.5	34.5	1.50
4	418	0.870	55.5	36.3	1.48
5	436	0.925	57.5	34.8	1.58
6	285	0.604	37.6	53.2	0.85
7	259	0.550	34.2	58.5	1.02
8	373	0.720	49.2	40.6	0.86
9	248	0.526	32.7	61.1	1.64
10	126	0.267	16.6	120.4	....
11	363	0.770	47.8	41.8	1.31
12	448	0.950	59.1	33.8	....
13	287	0.608	37.8	52.9	1.32
14	298	0.630	39.3	50.9	1.77

41 The original weight of test pieces and carbonizing material used in each pot were obtained before each run.

42 The American Gas Furnace Company's gas-fired muffle furnace No. 4 was used in case carbonizing. It was found possible to run six pots at a heat, equally spaced in muffle, not touching each other or the sides of furnace. Furnace-working temperatures were obtained by regular Le Chatelier type thermocouple, the hot end of which projected down between the pots in the middle of the muffle. Each of the six pots used in a run contained a different kind of carbonizing agent. Three consecutive runs were made on each kind, the material from the preceding run being used in each case except, of course, the

first run, which was on new raw material. New test pieces were used for each run of each kind of carbonizer.

43 In conducting the carbonizing runs, the furnace working conditions were maintained as nearly uniform in each as possible. Pots were arranged in the muffle in the afternoon ready for starting and at 7 a.m. the furnace was lighted. The temperature was worked up at a fairly rapid rate and very close to 9 a.m. it had reached 1750 deg. fahr. in all cases. This temperature was maintained uniformly throughout the day and up to 6 p.m. when the furnace was shut off and furnace and pots were allowed to cool slowly over night. Subsequent examination included:

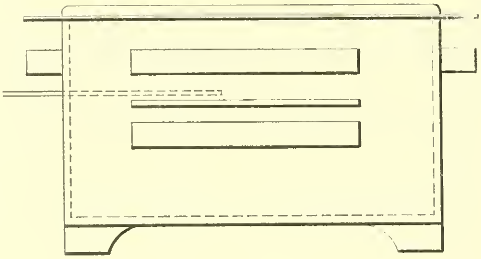

Fig. 1 Diagram showing Location of Test Pieces in Carbonizing Pot

- *a* Thorough examination of general appearance of pots, carbonizing material and test pieces after unpacking: weights obtained to show loss of weight of carbonizer, or gain of weight of test pieces, loss of volume of carbonizer. The mixtures were carefully saved and used without further rejuvenation, in packing the pots for the next run. Shrinkage in volume was taken care of by using sufficient sand to fill up pots. This was done by placing the added sand in the bottom of the pot.
- *b* Taking of cuts from solid cylinders for analysis. Bars after carbonizing, were carefully cleaned of all dirt, etc., set up on centers and tested for straightness or straightened, if necessary, until they ran absolutely true on centers. Then 11 consecutive cuts of 0.005 in. each on radius were taken, the length of the bar and turnings carefully kept separate and put in en-

TABLE 7 ANALYSIS OF CARBONIZING MATERIALS

Number of Material	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Price per Ton, Dollars and Cents	62	55	52	53.60	....	45	60	35	100	...	55	...	70	90
Approximate Analysis, Per Cent														
Moisture	5.65	5.57	6.50	5.90	7.03	2.4	1.70	2.26	4.60	4.10	1.55	0.45	6.15	0.85
Volatile Matter	31.17	33.73	32.90	33.14	30.87	25.77	10.73	12.43	10.90	15.95	32.35	23.10	54.95	15.53
Fixed Carbon	0.15	1.00	0.05	1.20	0.60	37.75	60.32	58.30	78.62	75.80	15.60	57.93	22.20	30.82
Ash	63.03	59.70	60.55	59.76	61.50	34.02	27.25	26.04	5.88	4.15	50.50	18.52	16.70	52.80
	100.00	100.00	100.00	100.00	100.00	99.94	100.00	99.03	100.00	100.00	100.00	100.00	100.00	100.00
Ash Analysis, Per Cent														
Calcium Oxide	34.31	32.27	not determined	not determined	33.49	17.00	7.95	6.64	0.52	0.67	26.17	8.39	6.68	....
Phosphorus Pentoxide	29.00	27.28			28.30	14.02	3.97	2.62	0.35	0.49	23.40	3.30	5.93	0.58
Aluminum Oxide	.....	.....			.....	1.82					0.76	0.52	0.28	
Iron Oxide	.....	.....			.....									
Silica	.....	.....			.....	0.58	5.76	5.46	0.11	0.08	0.34	0.79	0.69	1.58
Carbon Dioxide	.....	.....			.....	....	....	....	0.46	....	....	....	....	10.68
Sulphur Trioxide	.....	.....			.....	0.65	2.12	2.15	1.85	1.62	0.80	1.60	0.90	1.52
Sodium Oxide	.....	.....			.....	....	6.45 (by difference)	10.13 (by difference)	....	....	....	....	....	....
Magnesium Oxide	.....	.....			.....	....	....	....	1.43	1.18	....	3.29	0.63	0.68
Barium Oxide	.....	.....			.....	....	....	....	1.62	....	....	....	....	37.24
	100.28	99.85			100.29	100.05	99.00	100.00	100.46	99.89	100.97	99.37	98.41	99.48
Solid Matter and Carbon Dioxide, in the Carbonizer														
Total Combustible Carbon	12.51	14.07	15.86	14.88	13.50	53.42	66.24	65.88	78.41	77.66	41.42	70.11	40.02	44.93
Carbon Dioxide as Carbonates	3.51	2.95	not determined	not determined	3.43	0.62	4.62	4.35	1.20	0.75	1.18	7.36	0.89	11.63
Total Sulphur	0.16	0.10		....	not determined	0.26	1.03	0.97	0.74	0.65	0.24	0.64	0.36	0.61
Total Phosphorus	12.66	11.91		....		7.00	0.005	0.002	0.024	0.24	10.22	1.44	2.59	0.01
Sulphur as Sulphide	.....	.....	.....	....	.....	....	0.08	0.15	....	....	....	....	....	....
Hydro-Carbon Oils (soluble in gasoline)	.....	.....	.....	....	.....	....	....	....	....	....	23.82	....	....	....
Approximate Specific Gravity	0.954	0.774	0.930	0.88	0.92	0.60	0.79	0.55	0.52	0.27	0.77	0.95	0.61	0.63

TABLE 8 MEASUREMENTS IN MILLIMETERS OF ZONES AND CARBON PENETRATION OF TEST PIECES

SEE APPENDIX NO. 2

Case-Carbonizing Material No.	Zone of Excess Cementite	Eutectic Zone	Gradation Zone	Total Penetration of Carbon
RUN NO. 1				
1	0.455	0.455	0.650	1.560
2	0.450	0.650	0.748	1.848
3	.....	0.520	1.235	1.755
4	.....	0.975	1.300	2.275
5	0.780	0.520	0.975	2.275
6	.....	.....	1.235	1.235
7	0.325	0.520	0.975	1.920
8	0.455	0.520	0.975	1.950
9	1.235	0.423	0.780	2.438
10	.....	0.975	1.105	2.080
11	.....	0.195	1.300	1.495
12	.....	0.339	1.235	1.574
13	0.325	0.520	1.300	2.145
14	1.040	0.339	0.650	2.029
RUN NO. 2				
1	.....	0.715	1.001	1.716
2	.....	0.650	1.073	1.723
3	.....	0.520	1.300	1.820
4	.....	0.878	0.975	1.853
5	.....	0.780	0.975	1.755
6	.....	.....	0.975	0.975
7	.....	2.260	1.300	1.560
8	0.325	0.715	0.910	1.950
9	1.075	0.455	0.780	2.300
10	.....	0.065	1.170	1.235
11	.....	.....	1.658	1.658
12	.....	0.130	1.560	1.690
13	.....	0.520	1.170	1.690
14	.....	1.300	0.450	1.105
RUN NO. 3				
1	.....	0.358	1.138	1.496
2	.....	.....	0.975	0.975
3	.....	.....	0.325	0.325
4	.....	0.455	0.975	1.430
5	.....	0.225	1.300	1.525
6	.....	.....	0.040	0.040
7	.....	0.715	1.170	1.995
8	0.195	0.650	0.650	1.495
9	0.325	0.845	0.975	2.145
10	.....	.....	0.910*	0.975*
11	.....	.....	1.235†	1.242†
12	.....	0.007	1.560	1.567
13	.....	0.293	1.170	1.463
14	1.170	0.845	1.300	3.315

* On the outside of this specimen there was a rim of almost pure ferrite 0.065 mm. thick.

† On the outside of this specimen there was a rim of almost pure ferrite 0.007 mm. thick.

velopes for subsequent determinations of carbon, phosphorus and sulphur.

c Breaking of carbonized tube for general appearance and action of carbonizer in tubes, etc.

d Breaking of ½ in. round rod (some after heat treatment) for general appearance, depth of case, and microscopic examination.

44 In Table 6 are listed data concerning the 14 carbonizing materials and in Table 7 are given the analyses. In Figs. 2 to 15 are shown graphically the carbon contents of the carbonized sections of the test bars at successive depths, as explained in Par. 43 *b*. The captions of these diagrams give also the phosphorous and sulphur contents in each case at depths of 0.005 in. and 0.06 in. below the surface of the test bars.

45 In Appendix No. 3 is a detailed description of the results of the microscopic examination of the several bars and in Table 8 is summarized the measurements in millimeters, determined by the microscopic examination, of the zone of excess cementite, the eutectic zone, the gradation zone and the depth of carbon penetration.

46 In Table 9 are summarized physical and price data on the carbonizing materials in relation to the carbon penetration of the test bars.

47 The results of the experiment on the efficiency of case-carbonizing materials may be summarized as follows:

a The volume per ton of case-carbonizing materials varies. As these materials are used by volume, not by weight, this volume per ton must be considered in the economic purchase of this grade of material.

b All the compounds show shrinkage of volume and in most instances a loss of potency with continued service. This point is of greatest commercial interest since, if steel of variable carbon content is produced in the case-hardening process the manufacturer cannot hope for uniform results.

c With the compounds on the market, any desired maximum carbon content in the case can be obtained with a given temperature simply by varying the compounds used.

d When parts are produced which require grinding after

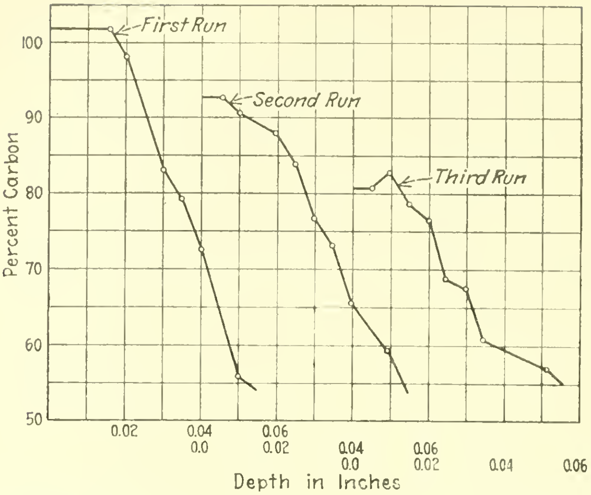

FIG. 2 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 1

PHOSPHORUS CONTENT 0.025%-0.029%; SULPHUR CONTENT 0.04%-0.04%

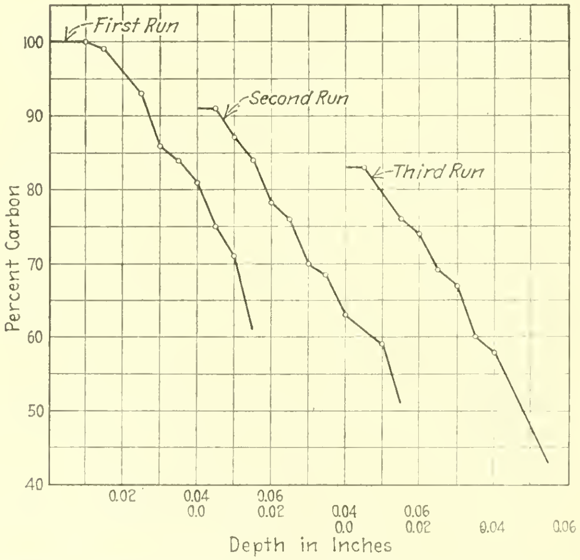

FIG. 3 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 2

PHOSPHORUS CONTENT 0.030%-0.032%

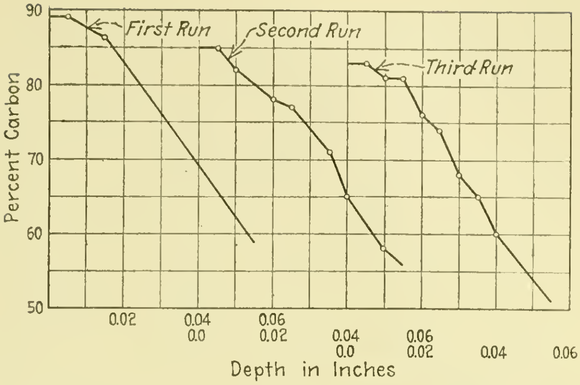

FIG. 4 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 3
PHOSPHORUS CONTENT 0.029%–0.033%

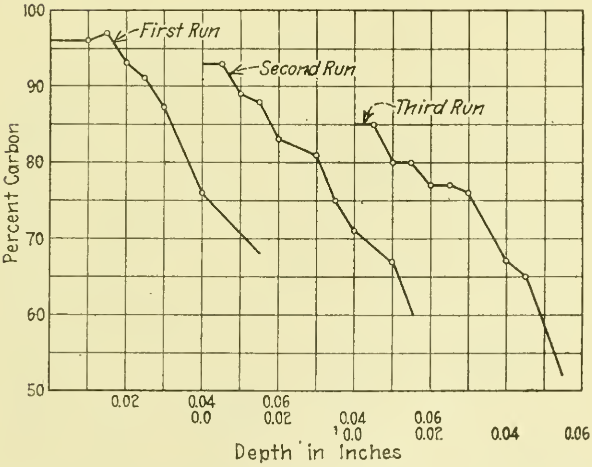

FIG. 5 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 4
PHOSPHORUS CONTENT 0.029%–0.036%

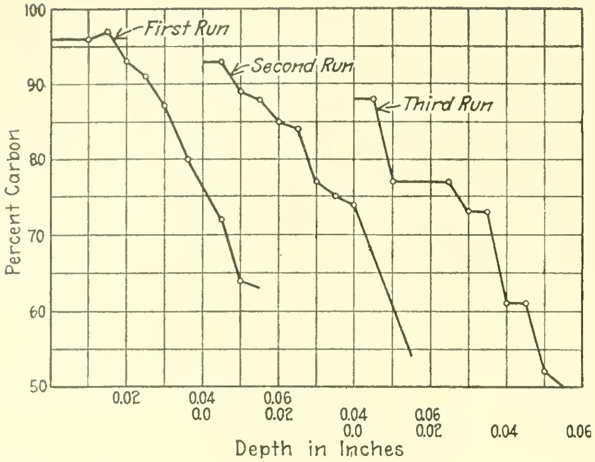

FIG. 6 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 5
PHOSPHORUS CONTENT 0.032%-0.038%

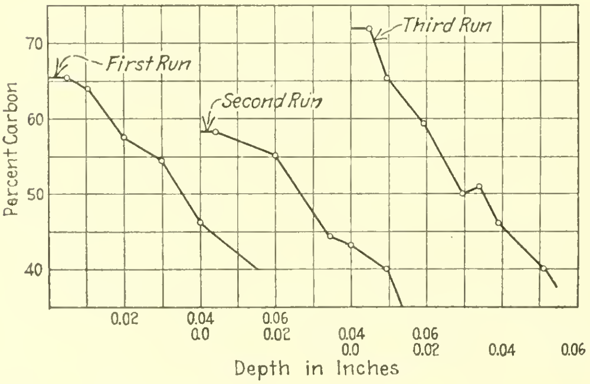

FIG. 7 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 6
PHOSPHORUS CONTENT 0.028%-0.030%; SULPHUR CONTENT 0.034%-0.040%

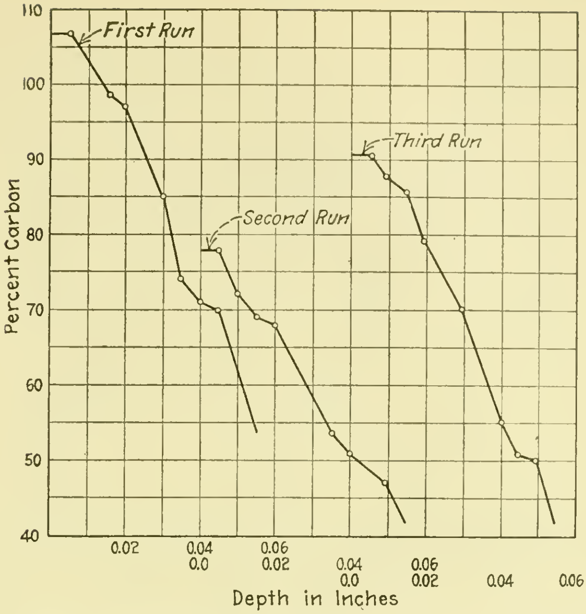

FIG. 8 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 7
PHOSPHORUS CONTENT 0.024%–0.033%; SULPHUR CONTENT 0.040%–0.044%

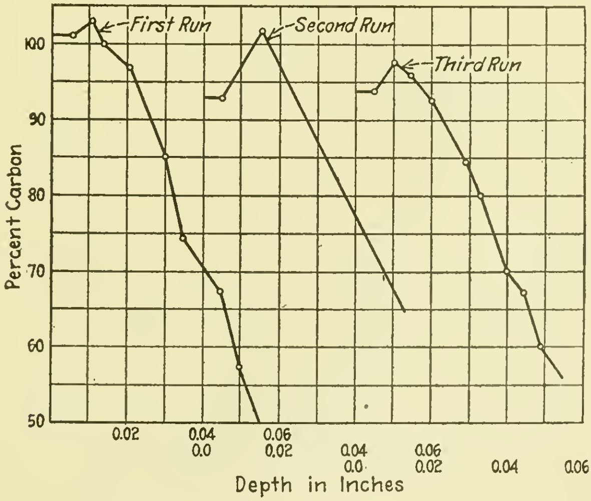

FIG. 9 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 8
PHOSPHORUS CONTENT 0.028%–0.031%; SULPHUR CONTENT 0.042%–0.046%

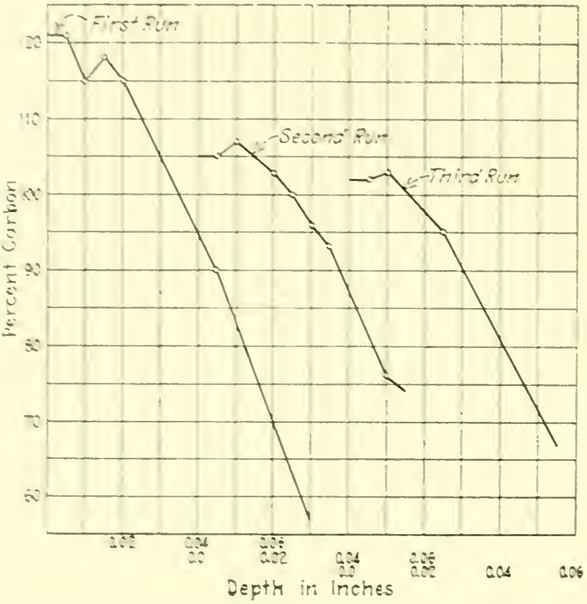

FIG. 10 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 9

PHOSPHORUS CONTENT 0.026%–0.036%: SULPHUR CONTENT 0.050%–0.054%

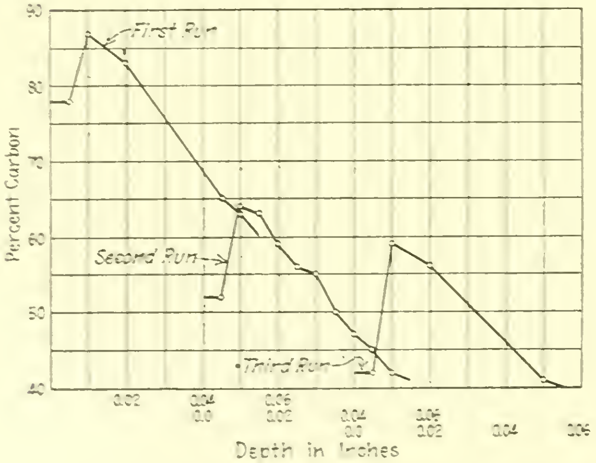

FIG. 11 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 10

PHOSPHORUS CONTENT 0.042%–0.035%: SULPHUR CONTENT 0.044%–0.042%

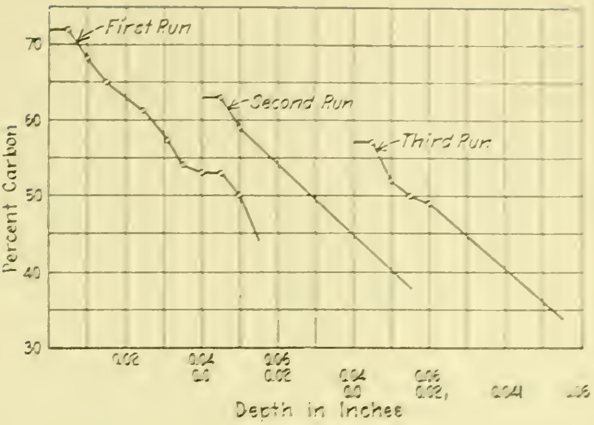

FIG. 12 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 11
PHOSPHORUS CONTENT 0.033%–0.036%; SULPHUR CONTENT 0.042%–0.042%

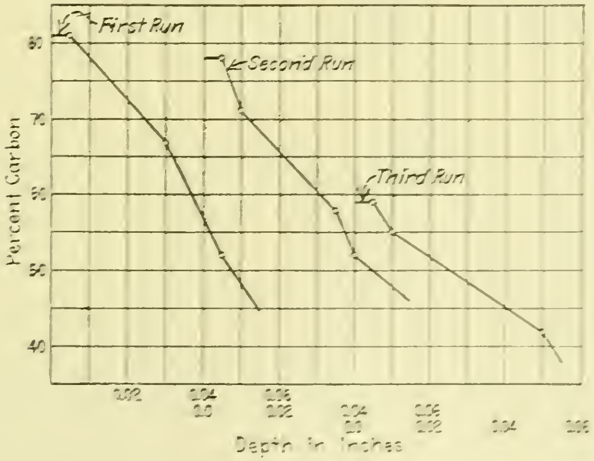

FIG. 13 CARBON PENETRATION CURVES FOR TEST BARS TREATED WITH CARBONIZING MATERIAL NO. 12
PHOSPHORUS CONTENT 0.031%–0.036%; SULPHUR CONTENT 0.044%–0.044%

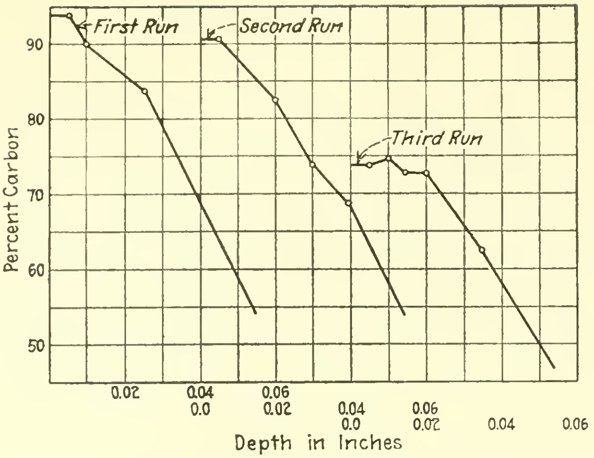

Fig. 14 Carbon Penetration Curves for Test Bars treated with Carbonizing Material No. 13
Phosphorus Content 0.072%–0.032%; Sulphur Content 0.044%–0.042%

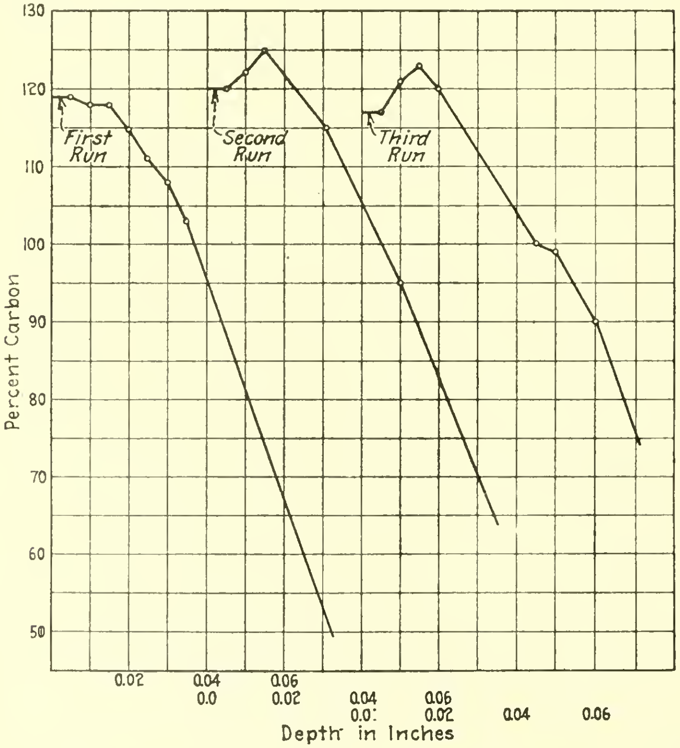

Fig. 15 Carbon Penetration Curves for Test Bars treated with Carbonizing Material No. 14
Phosphorus Content 0.036%–0.034%; Sulphur Content 0.036%–0.042%

TABLE 9 PHYSICAL AND PRICE DATA ON CARBONIZING MATERIALS IN RELATION TO CARBON PENETRATION

Carbonizing Material No.	Carbon in Test Bars, Per Cent						Relative C Yielded in Satisfactory Tests per Unit Volume	Price per Cu. Ft.	Cost per Cu. Ft per Unit
	Depth of 0.005 In.		Depth of 0.025 In.						
	1st	2d	3d	1st	2d	3d			
1	1.02	0.93	0.81	0.87	0.80	....	1.67	$1.84	$1.10
2	1.00	0.91	0.83	0.91	0.76	....	1.67	1.30	0.78
3	0.89	0.85	0.83	....	....	....	....	1.50	....
4	0.96	0.93	0.85	0.88	0.81	....	1.69	1.48	0.87
5	0.96	0.93	0.88	0.85	0.80	....	1.65	1.58	0.96
6	0.65	0.38	0.72	....	....	....	....	0.85	....
7	1.07	0.78	0.91	0.88	0.80	0.73	2.41	1.02	0.42
8	1.01	0.93	0.94	0.85	0.88	0.93	2.56	0.86	0.34
9	1.21	1.05	1.05	0.95	0.90	0.90	2.90	1.64	0.56
10	0.78	0.52	0.43	....	....	....	....	....	....
11	0.72	0.63	0.57	....	....	....	....	1.31	0.85
12	0.81	0.78	0.59	....	....	....	....	....	....
13	0.94	0.91	0.74	0.78	0.78	....	1.56	1.32	5.33
14	1.19	1.20	1.17	1.06	1.13	1.13	3.32	1.77	....

TABLE 10 THERMAL CONDUCTIVITY OF CARBONIZING COMPOUNDS

No.		Price per Ton	Weight per Cu. Ft.	Time to Heat to 1200° F., Min.	Per Cent of Fastest	Per Cent of Slowest
1	New case-carbonizing compound No. 8	$60	28	52	153	34.7
2	Old case-carbonizing compound No. 8	$60	28	34	100	22.7
3	New case-carbonizing compound No. 1	$57	52	145	425	96.5
4	Old case-carbonizing compound No. 1	$57	52	99	291	66.0
5	New case-carbonizing compound No. 14	$90	35	96	282	64.0
6	Old case-carbonizing compound No. 14	$90	35	110	324	73.5
7	New case-carbonizing compound No. 17	$40	51	114	336	76.0
8	Old case-carbonizing compound No. 17	$40	51	76	223	50.5
9	Case-carbonizing compound No. 15	$80	45	112	330	74.5
10	Case-carbonizing compound No. 11	$40	..	143	424	95.5
11	Case-carbonizing compound No. 16	...	..	150	442	100.0

treatment, a compound should be selected which gives a high maximum carbon content.

e When parts are produced which require resistance to shock after heat treatment, a compound should be selected which gives a carbon maximum sufficient for the hardness desired and not in excess of this amount, or brittleness will result.

f The ideal case-hardening compound, as indicated by the experiment, would possess the following characteristics: (1) large volume per ton; (2) small shrinkage per run; (3) high resistivity to change of shape or powdering; (4) cleanliness and freedom from dust; (5) uniform case-carbonizing power at all runs; (6) capability of being used an innumerable number of times.

48 These specifications may seem impossible for any compound. There is now on the market, however, a material which can be used over and over again without loss of carbonizing power, by the addition of new material, which fulfils the conditions mentioned.

EXPERIMENT NO. 6

EXPERIMENT OF THE THERMAL CONDUCTIVITY OF CARBONIZING MATERIALS

49 The object of this experiment was to determine the relative rapidity with which different carbonizing materials would heat up. This factor is of great practical importance in determining the length of time required for the carbonizing operation, inasmuch as previous experiments have shown that, using standard round carbonizing pot 9 in. in diameter and 15 in. high and carbonizing material No. 1, more than two-thirds of the total time needed to attain a depth of 1-32 in. of case under regular carbonizing practice is required in heating up the pot and contents to the carbonizing temperature. It will readily be seen that a saving in time in bringing the contents of the pot up to heat will materially decrease the cost of carbonizing operation.

50 The materials used in conducting this experiment were case-carbonizing materials Nos. 1, 8, 11, 14, 15 and 16. The materials were obtained as samples from the various manufac-

turers. In carrying out the experiment the plan was to heat a lead bath to a constant temperature and to record the time for a thermo-couple in a special pot to come up to some definite temperature. The pot was packed with the different carbonizing mediums. The lead pot was 24 in. long, 15 in. wide, and 7 in. deep, and was filled with lead to a depth so that when the pot containing the carbonizer was immersed in it, the lead stood within ½ in. of the top. The temperature of the lead pot was recorded by a calibrated Le Chatelier thermo-couple. The lead pot was covered by a sheet of steel ¼ in. thick, and containing holes to accommodate the carbonizer pot and thermocouple tube. The pot containing the carbonizer was made of 6 in. round pipe, ¼ in. wall, and 8½ in. high. A plug ¼ in. thick was welded in the bottom. Angle-iron handles were riveted to opposite sides by means of which the pot was held down in the lead bath. The handles were so located that they held the pot immersed to a depth of 5½ in. in the lead. The temperature of the inside of the pot was recorded by a calibrated Le Chatelier thermo-couple so located that the hot junction was equidistant from the bottom and sides of the pot. Both thermocouples were connected up to read on the H. H. Franklin Company's single pivot galvanometer through a double throw switch. The temperature of the cold junction of the couples was recorded.

51 In packing the pot with the different carbonizing materials it was filled up level full, after locating the thermo-couple properly, and then tamped in fairly tight. The pot was covered with a thin sheet steel cover containing a hole for the thermocouple tube and this cover was luted on the clay. Records were taken of the time of putting the packed pot in the lead and every five minutes thereafter of the pot and lead temperatures until the former reached 1200 deg. fahr. The recorded temperatures were corrected for cold junction temperature and the calibration correction and time temperature curves were drawn for all the compounds used.

52 The results of this experiment are summarized in Figs. 16 and 17, which give the numbers of the compounds upon which results are recorded, the time required to heat to 1200 deg. with each compound and the rapidity of rise in temperature shown by the thermo-couple packed in the different carbonizing compounds.

CONCLUSIONS

53 This experiment has served to place the different case-carbonizing materials in a definite order in regard to their rapidity of heating up to case-carbonizing temperatures. The thermal conductivity of case-carbonizing materials must be carefully considered when low cost for this operation is essential and when large section pots are used. The lag in pots of large section is often sufficient to produce very un-uniform penetration and thus cause greatest warpage and physical variation.

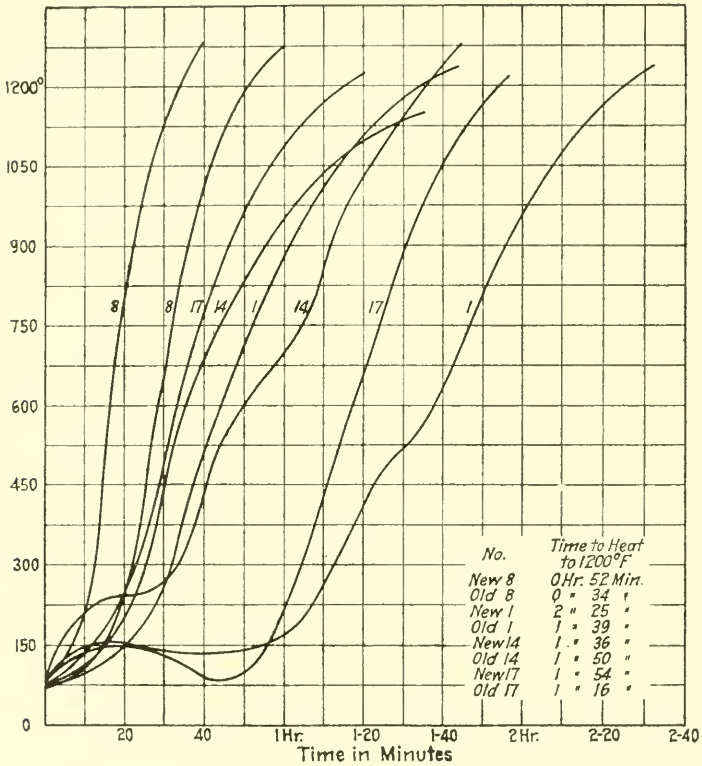

Fig. 16 Thermal Conductivity of Carbonizing Compounds

EXPERIMENT NO. 7

EFFICIENCY OF COPPER PLATING IN PREVENTING CARBONIZATION: GENERAL METHOD

54 The object of the experiment was to determine the efficiency of different thicknesses of copper plate in preventing absorption of carbon, and to ascertain the condition under which a plate could be obtained effectually to prevent carbonization.

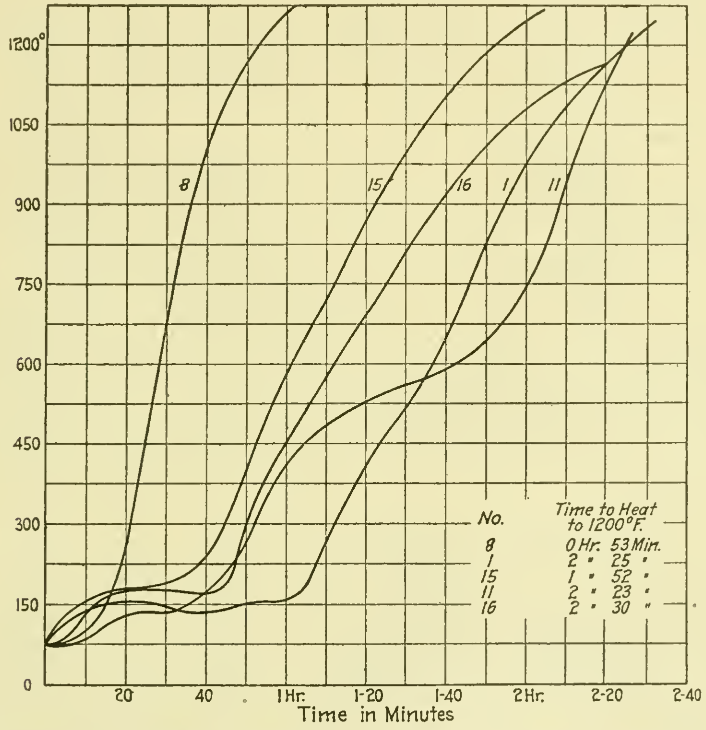

Fig. 17 Thermal Conductivity of Carbonizing Compounds

55 The stock used in the test pieces was steel No. 3 (0.20 per cent carbon basic open-hearth steel). The test pieces were 3½ in. long by 0.450 in. in diameter. They were set up on centers in a lathe and turned to 0.460 in. They were then ground to 0.450 in. and micrometered carefully to 1-10,000 in. All micrometer measurements were made by the same man.

56 The pieces were then copper plated, all conditions of the operation being kept constant except the actual time of plating,

which was carefully noted in each case. After plating the test pieces were micrometered and the thickness of the plate determined. The pieces were then carbonized under standard condition to give a depth of case of 1-32 in. Carbonizing furnace No. 2 was used and the material was case-carbonizing material No. 1, one-half new and one-half used. The pieces were then set up in a lathe and the copper plate removed with emery cloth, after which they were carefully micrometered again.

TABLE 11 MEASUREMENT[1] OF TEST PIECES; DURATION OF PLATING OPERATION; PER CENT CARBON ABSORBED

No.	Diameter before Copper Plating	Second Length Time in Plating Bath, Sec.	Diameter after Copper Plate	Thickness of Copper Plating
1	0.4496	about 1	0.4497	0.00005
2	0.4500	10	0.4501	0.00005
3	0.4500–0.4498	20	0.4501–0.4499	0.00005
4	0.4502	30	0.4503	0.00005
5	0.4501	60	0.4504	0.00015
6	0.4496	2	0.4500	0.0002
7	0.4500–0.4502	3	0.4507–0.4508	0.0003
8	0.4500	4	0.4509	0.00045
9	0.4499	5	0.4512–0.4509	0.00065–0.0005
10	0.4500	10	0.4525–0.4516	0.00125–0.0008

No.	Diameter after Removal of Copper	Thickness Metal Removed on Radius	Per Cent C Cut No. 1	Per Cent C Cut No. 2
1	0.450 (?)	....	0.96	0.95
2	0.4504–0.450 (?)	....	0.58	0.56
3	0.4499–0.4498	0.0002–0.0001	0.40	0.38
4	0.450	0.0003	0.46	0.42
5	0.4498	0.0006	0.20	0.18
6	0.4492–0.4490	0.0008–0.001	0.19	0.17
7	0.4496–0.4498	0.0013–0.001	0.18	0.16
8	0.4493–0.4494	0.0016–0.0015	0.20	0.18
9	0.4494–0.4492	0.0018–0.0016	0.17	0.16
10	0.4495–0.4490	0.003 –0.0026	0.17	0.16

57 Two cuts were taken on each test piece, each cut being 0.005 in. on the radius, reducing the diameter 0.010 in. The turnings obtained were carefully taken and were very fine. Carbon determinations by direct combustion were made on each cut.

58 Direct electric current for all plating operations was

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[1] All measurements made by one man and as carefully as possible with a 0.0001 in. micrometer

furnished by a 4-pole direct-current dynamo which is belt driven by an alternating-current constant speed motor. The rated capacity of the dynamo is 360 amperes at 6 volts.

Voltage across terminals.. .... 4.1
Temperature of bath.......................... 170 deg. fahr.

Operation No.	Contents of Bath	Purpose
1	Gasolene	To remove grease
2	Sawdust	To dry
3	Warm potassium hydroxide solution	To remove grease and dirt
4	Warm water	To wash
5	Warm sulphuric acid solution	To acid clean
6	Warm water	To wash
7	Cold water	Additional wash
8	Cold potassium cyanide solution	Cleanser
9	Cold water	To wash
10	Electric cleaner, warm sodium hydroxide case-iron anode	Cleanser to give good plating surface
11	Copper plating bath of copper sulphate and potassium cyanide solution warm	Plating bath

59 The copper cover to prevent the penetration entirely must have a thickness of 0.0005 in. Microscopic inspection before and after case carbonizing of copper plated pieces of steel showed that the copper coating was not continuous when the thickness was less than 0.0005 in.

GENERAL CONCLUSIONS

60 This paper may lack interest for the more scientific reader on account of the non-technical nature of the notes, and the want of timely information on the subject of case carbonizing. The information herein presented was obtained with the object of pointing out: (*a*) the quantitative effect of the commercial irregularities which creep into case-carbonized steel parts; (*b*) the precautions which must be observed if uniformity is desired; and (*c*) the physical characteristics in case-carbonizing materials which must be considered if low cost for this operation is to be obtained.

61 The ideal case-carbonizing compound perhaps has not yet been produced; however, compounds which are clean, mechanically strong, uniform in case-carbonizing power, and capable of unlimited re-use are being commercially employed in case-carbonizing operation. This is substantial progress.

ACKNOWLEDGMENTS

62 The researches are the results of several years' study and the product of the efforts of more than one man. Messrs. Fenner, Baxter, Nead and Kiefer, members of the metallurgical staff of the H. H. Franklin Manufacturing Company, Mr. Bourg, metallurgist for the Brown Lipe Chapin Company, and Messrs. H. J. Stagg and C. R. Bulley of the author's staff in the metallurgical department of the Halcomb Steel Company have all contributed their share to the results herein reported and it was only by the coöperation of all that the vast amount of work necessary could be accomplished in so short a time. The author wishes, therefore, to thank these gentlemen and to credit them with their share in the work here reported. The author also wishes to thank Dr. Mathews, manager of the Halcomb Steel Company, for his advice and support. The companies served by the compilers of these data deserve recognition for their progressiveness and their contribution to the general knowledge of the subject in permitting the publication of this report. The subjects of interest connected with the commercial use of steels are so large that it is only by such coöperation that a great deal can be accomplished in a few years.

APPENDIX NO. 1

TEMPERATURES AT WHICH TEST PIECES WERE REHEATED OR ANNEALED, EXPERIMENT NO. 1

TABLE 12 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 2

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Quenched from pot at 1500° into water............	0.500	900	0.40	0.04
2	Reheat 1400°, quench in water....................	0.500	775	0.30	0.04
3	Reheat 1500°, quench in water....................	0.500	740	0.23	0.04
4	Reheat 1600°, quench in water....................	0.500	850	0.24	0.04
5	Reheat 1700°, quench in water....................	0.500	800	0.24	0.04
6	Reheat 1800°, quench in water....................	0.500	815	0.24	0.04
7	Reheat 1600°, quench oil, reheat 1400°, quench water	0.499	750	0.29	0.04
8	Reheat 1600°, quench oil, reheat 1500°, quench oil..	0.499	1050	0.50	0.04
9	Reheat 1600°, quench oil, reheat 1500°, quench water	0.500	1000	0.50	0.04
10	Reheat 1500°, quench water, draw 380° for 20 min..	0.501	820	0.24	0.04
11	Anneal 1400°, reheat 1500°, quench water..........	0.500	830	0.30	0.04
12	Anneal 1600°, quench oil, reheat 1500°, quench water	0.499	900	0.37	0.04

SUMMARY: 1, Best single heat treatment reheat 1600°, quench water; 2, best double heat treatment reheat 1600°, quench oil, reheat 1500°, quench oil; 3, best double heat treatment increases strength 23 per cent and increases toughness 105 per cent; 4, drawing the temper increases strength 10 per cent and toughness 3 per cent; 5, annealing increases (wirh single treatment) strength 12 per cent, toughness 25 per cent; 6, annealing decreases (with double treatment) strength 10 per cent, toughness 26 per cent.

TABLE 13 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 3

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Reheat 1500°, quench in water	0.500	1020	0.61	0.010
2	Reheat 1500°, quench in water	0.500	950	0.44	0.020
3	Reheat 1500°, quench in water	0.500	940	0.29	0.025
4	Reheat 1500°, quench in water	0.500	850	0.31	0.030
5	Reheat 1500°, quench in water	0.500	800	0.30	0.035
6	Reheat 1500°, quench in water	0.500	725	0.24	0.040
7	Quenched from pot at 1500°, in water	0.501	940	0.30	0.040
8	Reheat 1400°, quench in water	0.501	700	0.21	0.040
9	Reheat 1500°, quench in water	0.500	700	0.22	0.040
10	Reheat 1600°, quench in water	0.500	810	0.24	0.040
11	Reheat 1700°, quench in water	0.500	810	0.25	0.040
12	Reheat 1800°, quench in water	0.500	810	0.24	0.040
13	Reheat 1600°, quench oil, reheat 1400°, quench water	0.499	840	0.30	0.040
14	Reheat 1600°, quench oil, reheat 1500°, quench oil	0.498	1000	0.39	0.040
15	Reheat 1600°, quench oil, reheat 1500°, quench water	0.499	970	0.36	0.040
16	Reheat 1500°, quench water, draw 380° for 20 min.	0.500	870	0.26	0.040
17	Anneal 1400°, reheat 1500°, quench water	0.500	845	0.26	0.040
18	Anneal 1600°, reheat 1600°, quench oil, reheat 1500°, quench water	0.500	830	0.26	0.040

Summary: 1, Strength and toughness decrease with increasing depth of case; 2, minimum strength and toughness with 0.040 in. depth of case; 3, best single heat treatment, reheat 1600 deg. fahr., quench water; 4, best double heat treatment, reheat 1600 deg. fahr., quench oil, reheat 1500 deg. fahr., quench oil; 5, double heat treatment increases strength 25 per cent, toughness 56 per cent; 6, drawing temper at 380 deg. fahr. increases strength 21 per cent, toughness 4 per cent; 7, annealing and single treatment increases strength 16.5 per cent, toughness 4 per cent; 8, annealing and double treatment decreases strength 14.5 per cent, toughness 29 per cent.

TABLE 14 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 4

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Reheat 1500°, quench water	0.500	830	0.48	0.010
2	Reheat 1500°, quench water	0.500	820	0.43	0.015
3	Reheat 1500°, quench water	0.500	740	0.35	0.020
4	Reheat 1500°, quench water	0.500	745	0.34	0.025
5	Reheat 1500°, quench water	0.500	725	0.32	0.030
6	Reheat 1500°, quench water	0.500	700	0.22	0.040
7	Quenched from pot at 1500° in water	0.500	810	0.24	0.040
8	Reheat 1400°, quench water	0.500	670	0.22	0.040
9	Reheat 1500°, quench water	0.500	700	0.22	0.040
10	Reheat 1600°, quench water	0.501	850	0.25	0.040
11	Reheat 1700°, quench water	0.500	830	0.25	0.040
12	Reheat 1800°, quench water	0.500	810	0.24	0.040
13	Reheat 1600°, quench oil, reheat 1400°, quench water	0.500	840	0.30	0.040
14	Reheat 1600°, quench oil, reheat 1500°, quench water	0.500	950	0.45	0.040
15	Reheat 1500°, quench water, draw 380° for 20 min.	0.500	790	0.26	0.040
16	Reheat 1600°, quench oil, reheat 1500°, quench water, draw 380° for 20 min.	0.500	950	0.44	0.040
17	Anneal 1400°, reheat 1500°, quench water	0.500	800	0.26	0.040

Summary: 1, Strength and toughness decrease with increasing depth of case; 2, minimum strength and toughness occur with depth of case of 0.020 in.; 3, best single heat treatment reheat 1600 deg. fahr., quench water; 4, best double heat treatment reheat 1600 deg. fahr., quench oil, reheat 1500 deg. fahr., quench water; 5, double heat treatment increases strength 11.8 per cent, toughness 80 per cent; 6, drawing increases with single treatment strength 12.9 per cent, toughness 18.2 per cent; 7, drawing increases with double treatment strength 0 per cent, toughness 0 per cent; 8, annealing increases with single treatment strength 14.3 per cent, toughness 18.2 per cent.

TABLE 15 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 5

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Reheat 1550°, quench oil	0.508	1225	2.66	0.014
2	Reheat 1550°, quench oil	0.508	1220	2.30	0.020
3	Reheat 1550°, quench oil	0.509	1215	1.75	0.025
4	Reheat 1550°, quench oil	0.509	1360	1.20	0.030
5	Reheat 1550°, quench oil	0.508	1295	0.70	0.035
6	Reheat 1550°, quench oil	0.509	1255	0.46	0.040
7	Quenched from pot at 1500° into oil	0.505	1400	0.53	0.040
8	Reheat 1310°, quench oil	0.504	850	0.46	0.040
9	Reheat 1400°, quench oil	0.504	1150	0.46	0.040
10	Reheat 1500°, quench oil	0.504	1250	0.46	0.040
11	Reheat 1600°, quench oil	0.504	1250	0.45	0.040
12	Reheat 1700°, quench oil	0.504	1280	0.46	0.040
13	Reheat 1800°, quench oil	0.504	1250	0.43	0.040
14	Reheat 1600°, quench oil, reheat 1400°, quench oil	0.504	1210	0.60	0.040
15	Reheat 1600°, quench oil, reheat 1400°, quench water	0.502	1200	0.42	0.040
16	Reheat 1600°, quench oil, reheat 1500°, quench oil	0.503	1500	0.51	0.040
17	Reheat 1600°, quench oil, reheat 1500°, quench water	0.504	1240	0.44	0.040
18	Reheat 1500°, quench oil, draw 380° for 20 min	0.504	1560	0.56	0.040
19	Reheat 1600°, quench oil, reheat 1400°, draw 380° for 20 min	0.504	1320	0.72	0.040
20	Anneal 1500°, reheat 1500°, quench oil	0.504	1280	0.43	0.040
21	Reheat 1500°, reheat 1600°, quench oil, reheat 1400°, quench oil	0.505	1200	0.60	0.040

SUMMARY: 1, Strength and toughness decrease with increase depth of case; 2, minimum strength and toughness occur with depth of case 0.040 in.; 3, best single heat treatment reheat 1500°, quench oil; 4, best double heat treatment reheat 1600°, quench oil, reheat 1500°, quench oil; 5, double heat treatment increases strength 20 per cent, toughness 10.8 per cent; 6, drawing increases with single heat treatment strength 24.8 per cent, toughness 10.8 per cent; 7, drawing increases with double heat treatment strength 9.1 per cent, toughness 20 per cent; 8, annealing increases with single heat treatment strength 2.4 per cent, toughness 6.5 per cent; 9, annealing increases with double heat treatment strength 0 per cent, toughness 0 per cent.

TABLE 16 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 6

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Reheat 1500°, quench oil	0.504	2010	2.40	0.010
2	Reheat 1500°, quench oil	0.504	1675	0.79	0.015
3	Reheat 1500°, quench oil	0.505	1615	0.69	0.020
4	Reheat 1500°, quench oil	0.505	1455	0.55	0.030
5	Reheat 1500°, quench oil	0.505	1390	0.52	0.035
6	Reheat 1500°, quench oil	0.504	1250	0.43	0.040
7	Quenched from pot at 1500° into oil	0.504	1350	0.49	0.040
8	Reheat 1300°, quench oil	0.503	880	0.32	0.040
9	Reheat 1400°, quench oil	0.503	1230	0.42	0.040
10	Reheat 1500°, quench oil	0.504	1315	0.45	0.040
11	Reheat 1600°, quench oil	0.504	1300	0.45	0.040
12	Reheat 1700°, quench oil	0.502	1320	0.45	0.040
13	Reheat 1800°, quench oil	0.502	1300	0.45	0.040
14	Reheat 1600°, quench oil, reheat 1300°, quench oil	0.503	1240	0.75	0.040
15	Reheat 1600°, quench oil, reheat 1400°, quench oil	0.505	1260	0.59	0.040
16	Reheat 1600°, quench oil, reheat 1400°, quench water	0.504	1200	0.37	0.040
17	Reheat 1600°, quench oil, reheat 1500°, quench oil	0.504	1390	0.44	0.040
18	Reheat 1600°, quench oil, reheat 1500°, quench water	0.504	1415	0.46	0.040
19	Reheat 1500°, quench oil, draw 380° for 20 min.	0.504	1610	0.53	0.040
20	Reheat 1600°, quench oil, reheat 1400°, quench oil, draw 380° for 20 min.	0.504	1400	0.65	0.040
21	Anneal 1500°, quench oil	0.504	1370	0.44	0 040
22	Anneal 1600°, quench oil, reheat 1400°, quench oil	0.504	1250	0.60	0 040

SUMMARY: 1, Strength and toughness decreases with increasing depth of case; 2, minimum strength and toughness occurs with depth of case of 0 010 in.; 3, best single heat treatment reheat 1500 deg. fahr., quench in oil; 4, best double heat treatment reheat at 1600 deg. fahr., quench in oil, reheat 1500 deg. fahr., quench oil; 5, best double heat treatment increases strength 6.1 per cent and toughness 0 per cent; 6, drawing increases with single treatment strength 23 per cent and toughness 17.8 per cent; 7, drawing increases with double treatment strength 11 per cent and toughness 10.1 per cent; 8, annealing increases with single treatment strength 4.1 per cent and toughness 0 per cent; 9, annealing increases with double treatment strength 0 per cent and toughness 0 per cent.

TABLE 17 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 7

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Reheat 1500°, quench oil	0.505	1975	2.50	0.010
2	Reheat 1500°, quench oil	0.505	1940	1.25	0.020
3	Reheat 1500°, quench oil	0.505	1760	0.96	0.025
4	Reheat 1500°, quench oil	0.506	1640	0.70	0.030
5	Reheat 1500°, quench oil	0.504	1410	0.53	0.035
6	Reheat 1500°, quench oil	0.505	1275	0.44	0.040
7	Quench from pot at 1500° into oil	0.504	1250	0.42	0.040
8	Reheat 1300°, quench oil	0.501	1000	0.33	0.040
9	Reheat 1400°, quench oil	0.500	1250	0.40	0.040
10	Reheat 1500°, quench oil	0.502	1280	0.44	0.040
11	Reheat 1600°, quench oil	0.502	1280	0.44	0.040
12	Reheat 1700°, quench oil	0.499	1300	0.45	0.040
13	Reheat 1800°, quench oil	0.499	1225	0.42	0.040
14	Reheat 1600°, quench oil, reheat 1300°, quench oil	0.501	1200	0.60	0.040
15	Reheat 1600°, quench oil, reheat 1400°, quench oil	0.500	1320	0.50	0.040
16	Reheat 1600°, quench oil, reheat 1400°, quench water	0.500	1170	0.37	0.040
17	Reheat 1600°, quench oil, reheat 1500°, quench oil	0.500	1380	0.41	0.040
18	Reheat 1600°, quench oil, reheat 1500°, quench water	0.500	1250	0.40	0.040
19	Reheat 1500°, quench oil, draw 380° for 20 min	0.503	1630	0.53	0.040
20	Reheat 1600°, quench oil, reheat 1400°, quench oil, draw 380° for 20 min	0.502	1525	0.58	0.040
21	Anneal 1500°, reheat 1500°, quench oil	0.502	1300	0.40	0.040
22	Anneal 1500°, reheat 1600°, quench oil, reheat 1400°, quench oil	0.502	1300	0.47	0.040

SUMMARY: 1, Toughness and strength decrease with increasing depth of case; 2, minimum strength and toughness occurs with depth of case of 0.040 in.; 3, best single heat treatment reheat 1450 deg. fahr., quench oil; 4, best double heat treatment reheat 1600 deg. fahr., reheat 1400 deg. fahr., quench oil; 5, best double heat treatment increases strength 5.6 per cent and toughness 13.6 per cent; 6, drawing increases with single treatment, strength 27.4 per cent, and toughness 20.5 per cent; 7, drawing increases with double treatment, strength 15.5 per cent, and toughness 16 per cent; 8, annealing increases with single treatment, strength 4 per cent, and toughness 0 per cent; 9, annealing decreases with double treatment, strength 1.5 per cent, and toughness 6 per cent.

TABLE 18 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 8

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in In.	Depth of Case in in.
1	Reheat 1550°, quench oil........................	0.507	2055	1.14	0.010
2	Reheat 1550°, quench oil........................	0.508	1625	0.55	0.020
3	Reheat 1550°, quench oil........................	0.507	1375	0.43	0.040
4	Reheat 1550°, quench oil........................	0.507	1350	0.42	0.060
5	Reheat 1550°, quench oil........................	0.506	1350	0.42	0.070
6	Reheat 1550°, quench oil........................	0.506	1350	0.42	0.095
7	Quenched from pot at 1500° into oil...............	0.506	1350	0.45	0.040
8	Reheat 1300°, quench oil........................	0.503	800	0.28	0.040
9	Reheat 1400°, quench oil........................	0.504	1100	0.34	0.040
10	Reheat 1500°, quench oil........................	0.503	1250	0.40	0.040
11	Reheat 1600°, quench oil........................	0.504	1300	0.43	0.040
12	Reheat 1700°, quench oil........................	0.506	1360	0.44	0.040
13	Reheat 1800°, quench oil........................	0.504	1250	0.41	0.040
14	Reheat 1550°, quench oil, reheat 1300°, quench oil..	0.504	1300	0.48	0.040
15	Reheat 1550°, quench oil, reheat 1400°, quench oil..	0.505	1450	0.50	0.040
16	Reheat 1550°, quench oil, reheat 1400°, quench water	0.505	1400	0.39	0.040
17	Reheat 1550°, quench oil, reheat 1500°, quench oil..	0.504	1440	0.49	0.040
18	Reheat 1550°, quench oil, reheat 1500°, quench water	0.502	1340	0.37	0.040
19	Reheat 1550°, quench oil, draw 380° for 20 min....	0.503	1750	0.60	0.040
20	Reheat 1500°, quench oil, reheat 1400°, quench oil, draw 380° for 20 min..........................	0.505	1580	0.53	0.040
21	Anneal 1500°, reheat 1500°, quench oil............	0.504	1370	0.43	0.040
22	Anneal 1500°, reheat 1550°, quench oil, reheat 1400°, quench oil..................................	0.505	1440	0.46	0.040

SUMMARY: 1, Strength and toughness decrease with increase in depth of case; 2, minimum strength and toughness occur with depth of case 0.040 in.; 3, best single treatment reheat 1500°, quench oil; 4, best double treatment reheat 1550°, quench oil, reheat 1400°, quench oil; 5, best double treatment increases strength 6.9 per cent, toughness 19 per cent; 6, drawing increases with best single treatment, strength 29.6 per cent, toughness 43 per cent; 7, drawing increases with best double treatment 17 per cent, toughness 23.8 per cent; 8, annealing increases with best single treatment 1.5 per cent, toughness 2.4 per cent; 9, annealing decreases with best double treatment, strength 0.7 per cent, toughness 8 per cent.

TABLE 19 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 9

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Reheat 1500°, quench oil	0.508	1595	0.52	0.010
2	Reheat 1500°, quench oil	0.508	1450	0.46	0.020
3	Reheat 1500°, quench oil	0.509	1450	0.48	0.040
4	Reheat 1500°, quench oil	0.509	1455	0.45	0.060
5	Reheat 1500°, quench oil	0.508	1435	0.47	0.070
6	Reheat 1500°, quench oil	0.509	1400	0.44	0.095
7	Quenched from pot at 1500° into oil	0.504	1700	0.50	0.040
8	Reheat 1300°, quench oil	0.507	1075	0.44	0.040
9	Reheat 1400°, quench oil	0.507	1450	0.44	0.040
10	Reheat 1500°, quench oil	0.507	1450	0.47	0.040
11	Reheat 1600°, quench oil	0.504	1450	0.48	0.040
12	Reheat 1700°, quench oil	0.503	1500	0.50	0.040
13	Reheat 1800°, quench oil	0.503	1500	0.50	0.040
14	Reheat 1550°, quench oil, reheat 1300°, quench oil	0.504	1300	0.47	0.040
15	Reheat 1550°, quench oil, reheat 1400°, quench oil	0.505	1550	0.47	0.040
16	Reheat 1550°, quench oil, reheat 1400°, quench water	0.506	1170	0.32	0.040
17	Reheat 1550°, quench oil, reheat 1500°; quench oil	0.506	1450	0.46	0.040
18	Reheat 1550°, quench oil, reheat 1500°, quench water	0.503	1240	0.34	0.040
19	Reheat 1550°, quench oil, draw 380° for 20 min.	0.506	1950	0.60	0.040
20	Reheat 1560°, quench oil, reheat 1400°, quench oil, draw 380° for 20 min.	0.506	1950	0.62	0.040
21	Anneal 1500°, reheat 1500°, quench oil	0.506	1500	0.18	0.040
22	Anneal 1500°, reheat 1550°, quench oil, reheat 1400°, quench oil	0.506	1500	0.48	0.040

SUMMARY: 1, The strength and toughness decreases with increase in depth of case; 2, minimum strength and toughness occur with depth of case of 0.02 in.; 3, best single treatment reheat 1400°, quench oil; 4, best double treatment reheat 1500°, quench oil, reheat 1400°, quench oil; 5, best double treatment increases strength 6.9 per cent and toughness 0 per cent; 6, drawing increases with single treatment, strength 34.4 per cent, toughness 27.6 per cent; 7, drawing increases with double treatment strength 25.8 per cent, toughness 40.5 per cent; 8, annealing increases with single treatment, strength 4.1 per cent, toughness 2.1 per cent; 9, annealing decreases with double treatment, strength 3.3 per cent, toughness 0 per cent.

TABLE 20 TRANSVERSE TESTS OF CASE-CARBONIZED STEEL NO. 10

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Depth of Case in in.
1	Reheat 1700°, quench water	0.505	1700	1.04	0.105
2	Reheat 1700°, quench water	0.507	1630	0.85	0.020
3	Reheat 1700°, quench water	0.505	1600	0.72	0.025
4	Reheat 1700°, quench water	0.508	1550	0.54	0.030
5	Reheat 1700°, quench water	0.507	1410	0.42	0.035
6	Reheat 1700°, quench water	0.510	1385	0.42	0.040
7	Quenched from pot at 1700°	0.506	1490	0.49	0.040
8	Reheat 1400°, quench water	0.506	1150	0.53	0.040
9	Reheat 1500°, quench water	0.506	1200	0.40	0.040
10	Reheat 1600°, quench water	0.506	1400	0.40	0.040
11	Reheat 1700°, quench water	0.506	1400	0.42	0.040
12	Reheat 1800°, quench water	0.506	1390	0.41	0.040
13	Reheat 1700°, quench oil, reheat 1400°, quench water	0.504	1300	1.60	0.040
14	Reheat 1700°, quench oil, reheat 1500°, quench water	0.503	1240	0.38	0.040
15	Reheat 1700°, quench oil, reheat 1600°, quench oil	0.502	1550	0.67	0 040
16	Reheat 1700°, quench oil, reheat 1600°, quench water	0.504	1500	0.44	0.040
17	Reheat 1700°, quench oil, reheat 1700°, quench oil	0.504	1500	0.49	0.040
18	Reheat 1700°, quench oil, reheat 1700°, quench water	0.503	1450	0.48	0.040
19	Reheat 1700°, quench water, draw at 380° for 20 min.	0.504	1600	0.50	0 040
20	Reheat 1700°, quench oil, reheat 1600°, quench water, draw 380° for 20 min.	0.504	1750	0.50	0.040
21	Anneal 1550°, reheat 1700°, quench water	0.504	1440	0.43	0.040

SUMMARY: 1, Strength and toughness decrease with increase in depth of case; 2, minimum strength and toughness occur with depth of case of 0.035 per cent; 3, best single heat treatment reheat 1600 deg. fahr. to 1700 deg. fahr., quench water; 4, best double heat treatment reheat 1700 deg. fahr., quench oil, reheat 1600 deg. fahr., quench oil; 5, best double heat treatment increases strength 10.7 per cent, toughness 92.5 per cent; 6, drawing increases with single treatment, strength 14.3 per cent, toughness 25 per cent; 7, drawing increases with double treatment, strength 16.7 per cent, toughness 13.6 per cent; 8, annealing incresaes with single treatment, strength 2.8 per cent, toughness 7 per cent.

TABLE 21 TRANSVERSE TESTS ON OIL-HARDENED STEELS

STEEL NO. 11

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Hardness	Remarks
1	Heated 1700°, quench in oil, drawn 212° for 20 min..................	508	2000	0.75	576	....
2	Heated 1700°, quench in oil, drawn 400° for 20 min.................	508	3025	1.55	542	....
3	Heated 1700°, quench in oil, drawn 600° for 20 min.................	508	2375	2.50	508	Did not break

STEEL NO. 12

No.	Heat Treatment	Size in in.	Load in lb. at Rupture	Deflection of Center at Rupture in in.	Hardness	Remarks
1	Heated 1500°, quenched in oil, drawn 212° for 20 min..................	508	1900	0.60	562	....
2	Heated 1500°, quenched in oil, drawn 400 ° for 20 min.................	508	2825	1.50	545	....
3	Heated 1500°, quenched in oil, drawn 600° for 20 min.................	508	2470	2.50	513	Did not break

APPENDIX NO. 2

TEMPERATURE RECORDS FOR EXPERIMENT NO. 3

TABLE 22 TEMPERATURE RECORD RUN NO. 1, 1300 DEG. FAHR. MATERIAL: BAR 291-2 (0.20% CARBON OPEN HEARTH); BAR 726-2 (0.20% NICKEL); BAR 353-2 (0.20% "A" VANADIUM)

Time	Temperature, Deg. Fahr.				Remarks
	Cold Junction	Furnace	Pot	Pot Corrected	
9.17	90	100	atmos.	....	Furnace started
9.30	97	530	....	....	
9.45	98	880	....	....	
10.00	106	1035	....	....	
10.15	110	1240	490	475	
10.30	106	1400	870	855	
10.45	104	1335	1200	1185	
11.00	108	1320	1310	1295	Begin to count time
11.15	112	1220	1320	1300	
11.30	114	1160	1300	1280	
11.45	118	1225	1260	1240	
12.00	112	1320	1280	1260	
12.15	116	1355	1330	1310	
12.30	...	1360	1330	1310	
12.45	...	1280	1350	1320	
1.00	117	1285	1360	1340	
1.15	118	1255	1330	1310	
1.30	124	1290	1320	1295	
1.45	122	1300	1320	1295	
2.00	117	1255	1335	1315	
2.15	121	1225	1315	1295	
2.30	124	1225	1290	1265	
2.45	124	1345	1310	1280	
3.00	132	1340	1360	1330	
3.15	128	1260	1360	1335	
3.30	120	1200	1330	1305	
3.45	117	1200	1300	1280	
4.00	126	1250	1270	1245	Pot pulled

TABLE 23 TEMPERATURE RECORD RUN NO. 2, 1400 DEG. FAHR.

Time	Temperature, Deg. Fahr.				Remarks
	Cold Junction	Furnace	Pot	Pot Corrected	
8.55	95	400	....	....	Furnace started
9.05	98	500	....	....	
9.15	95	840	....	....	
9.30	100	1040	....	....	
9.45	103	1190	....	....	
10.00	105	1325	715	700	
10.15	105	1445	1150	1135	
10.30	110	1440	1330	1315	
10.45	110	1460	1440	1425	Begin to count time
11.00	112	1400	1460	1440	
11.15	116	1360	1440	1420	
11.30	119	1335	1415	1395	
11.45	114	1385	1410	1390	
12.00	125	1435	1440	1415	
12.15	123	1420	1465	1440	
12.30	120	1400	1460	1435	
12.45	120	1380	1420	1400	
1.00	120	1415	1415	1395	
1.15	125	1345	1400	1375	
1.30	124	1355	1400	1375	
1.45	130	1435	1425	1400	
2.00	130	1440	1450	1425	
2.15	125	1405	1445	1420	
2.30	126	1375	....	....	No. 4 couple off
2.45	...	1320	....	....	
3.00	128	1300	1310	1285	
3.15	132	1410	1325	1290	
3.30	125	1475	1400	1375	
3.45	120	1460	1430	1405	
4.00	125	1420	1425	1400	
4.15	125	1375	1405	1380	Pot pulled

TABLE 24 TEMPERATURE RECORD RUN NO. 3, 1500 DEG. FAHR. MATERIAL: BAR 292-2 (0.20% CARBON OPEN HEARTH); BAR 728-2 (0.20% NICKEL); BAR 354-2 (0.2?% "A" VANADIUM)

Time	Temperature, Deg. Fahr.				Remarks
	Cold Junction	Furnace	Pot	Pot Corrected	
8.40	...	150	....	....	Furnace started
9.00	98	650	....	....	
9.15	98	910	....	....	
9.30	101	1090	....	....	
9.45	101	1230	490	475	
10.00	103	1345	910	895	
10.15	106	1440	1190	1175	
10.30	110	1535	1385	1370	
10.45	111	1540	1520	1505	
11.00	112	1540	1570	1550	
11.15	120	1500	1535	1515	
11.30	121	1465	1565	1540	
11.45	118	1460	1525	1505	
12.00	116	1455	1545	1520	
12.15	120	1400	1520	1500	
12.30	120	1360	1280 (?)	1260 (?)	
12.45	120	1420	1400	1380	
1.00	120	1470	1420	1400	
1.15	120	1455	1555	1530	
1.30	126	1370	1530	1505	
1.45	121	1360	1475	1450	
2.00	128	1410	1460	1435	
2.15	132	1480	1525	1495	
2.30	125	1480	1580	1565	
2.45	124	1425	1545	1520	
3.00	128	1380	1510	1485	
3.15	130	1365	1500	1475	
3.30	128	1420	1475	1450	
3.45	122	1500	1520	1496	
4.00	120	1390	1570	1555	
4.15	127	1320	1500	1475	Pot pulled

TABLE 25 TEMPERATURE RECORD RUN NO. 4, 1600 DEG. FAHR. MATERIAL: BAR 293-1 (0.20% CARBON OPEN HEARTH); BAR 732-1 (0.20% NICKEL); BAR 355-1 (0.20% "A" VANADIUM)

Time	Temperature, Deg. Fahr.				Remarks
	Cold Junction	Furnace	Pot	Pot Corrected	
10.00	...	....	....	....	Furnace started
10.30	...	1000	....	....	
10.45	...	1210	....	....	
11.00	...	....	....	....	
11.15	80	1460	910	895	
11.30	85	1565	1235	1210	
11.45	86	1655	1465	1445	
12.00	88	1655	1600	1580	
12.15	...	....	....	....	
12.30	...	....	....	....	
12.45	...	....	....	....	
1.00	102	1515	1545	1520	
1.15	99	1565	1565	1540	
1.30	101	1530	1550	1525	
1.45	95	1550	1560	1435	
2.00	84	1625	1600	1580	
2.15	102	1650	1635	1610	
2.30	100	1640	1655	1625	
2.45	90	1580	1655	1625	
3.00	74	1585	1640	1675	
3.15	92	1560	1600	1580	
3.30	90	1555	1500	1475	
3.45	105	1540	1565	1540	
4.00	92	1620	1600	1580	
4.15	...	....	....	....	
4.30	96	1655	1650	1625	
4.45	98	1585	1640	1615	
5.00	98	1530	1595	1575	Pot pulled

TABLE 26 TEMPERATURE RECORD RUN NO. 5, 1700 DEG. FAHR. MATERIAL: BAR 293-2 (0.20% CARBON OPEN HEARTH); BAR 732-2 (0.20% NICKEL); BAR 355-2 (0.20% "A" VANADIUM)

Time	Temperature, Deg. Fahr.				Remarks
	Cold Junction	Furnace	Pot	Pot Corrected	
8.45	...	....	....	....	Furnace started
9.00	...	....	....	....	
9.15	...	1100	....	....	
9.30	...	1300	....	....	
9.45	74	1405	500	980	
10.00	79	1570	980	955	
10.15	76	1700	1375	1330	
10.30	78	1740	1565	1540	
10.45	90	1600	1635	1610	
11.00	88	1590	1610	1590	
11.15	88	1585	1600	1580	
11.30	92	1695	1630	1605	
11.45	102	1800	1740	1725	
12.00	100	1740	1750	1735	
12.15	108	1700	1740	1725	
12.30	...	....	....	....	
12.45	...	....	....	....	
1.00	...	....	....	....	
1.15	91	1650	1665	1650	
1.30	85	1650	1675	1660	
1.45	90	1715	1715	1700	
2.00	91	1685	1720	1705	
2.15	91	1660	1685	1670	
2.30	110	1665	1680	1665	
2.45	90	1665	1685	1670	
3.00	90	1665	1685	1670	
3.15	88	1675	1700	1685	
3.30	104	1675	1680	1665	
3.45	103	1675	1690	1675	
4.00	...	1685	1700	1685	
4.15	100	1695	1705	1690	
4.30	98	1695	1705	1690	
4.45	90	1700	1715	1700	Pot pulled

TABLE 27 TEMPERATURE RECORD RUN NO. 26, 1800 DEG. FAHR. MATERIAL: BAR 284–1 (0.20% CARBON OPEN HEARTH); BAR 469–1 (0.20% NICKEL); BAR 455–1 (0.20% "A" VANADIUM)

Time	Temperature, Deg. Fahr.				Remarks
	Cold Junction	Furnace	Pot	Pot Corrected	
10.00	...	....	....	....	Furnace started
10.15	...	....	....	....	
10.30	...	....	....	....	
10.45	...	1220	....	....	
11.00	...	1380	....	....	
11.15	...	....	....	....	
11.30	88	1620	1215	1200	
11.45	96	1760	1500	1475	
12.00	97	1805	1710	1695	
12.15	109	1810	1780	1780	
12.30	...	....	....	....	
12.45	...	....	....	....	
1.00	108	1865	1870	1860	
1.15	112	1745	1835	1825	
1.30	104	1760	1780	1770	
1.45	106	1805	1810	1800	
2.00	106	1815	1825	1815	
2.15	96	1780	1820	1810	
2.30	...	....	....	....	
2.45	113	1775	1800	1790	
3.00	115	1780	1800	1790	
3.15	105	1780	1805	1795	
3.30	111	1780	1800	1790	
3.45	115	1785	1800	1790	
4.00	102	1785	1805	1795	
4.15	105	1780	1800	1790	
4.30	112	1775	1800	1790	
4.45	101	1770	1795	1785	
5.00	116	1765	1786	1775	Pot pulled

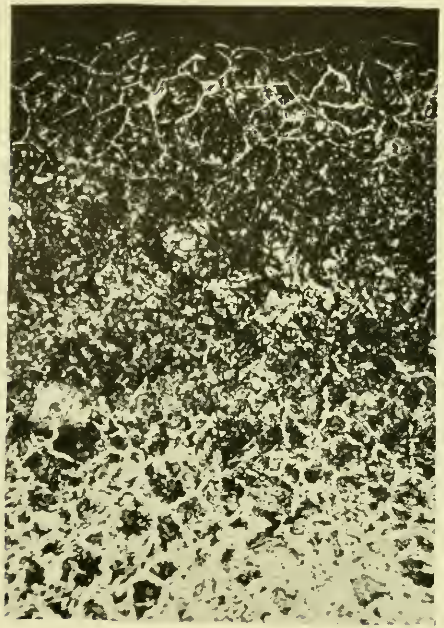

Fig. 18 Open Hearth Steel, 0.20% Carbon, Case Carbonized at 1700 Deg. Fahr. Shows Entire Zone affected by Case-Carbonizing Operation; 50 Diameters

In comparison with Figs. 28 and 29 note: (*a*) relatively small carbon penetration; (*b*) mass of cementite needles in outer zone.

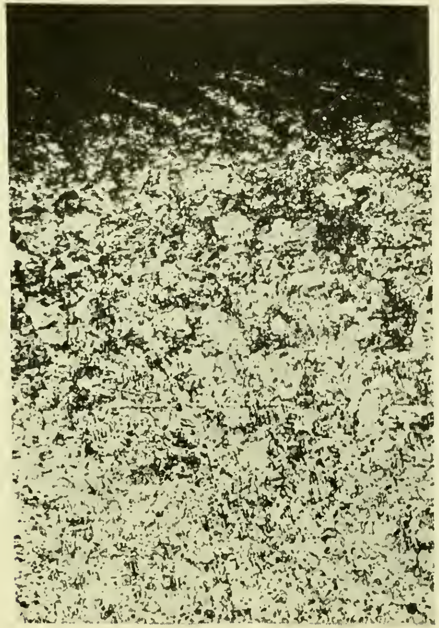

Fig. 19 Open Hearth Steel, 0.20% Carbon, Case Carbonized at 1300 Deg. Fahr. Shows Ferrite Band penetrating Carbonized Zone; 50 Diameters

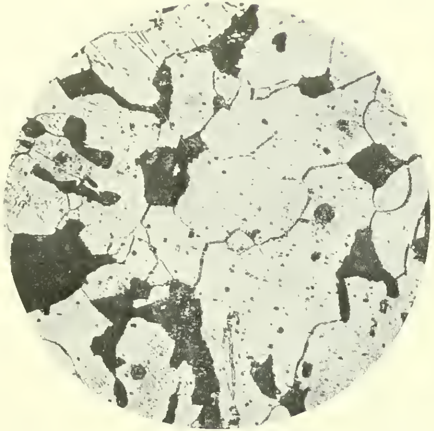

Fig. 20 Open Hearth Steel, 0.20% Carbon, Case Carbonized at 1700 Deg. Fahr. Structure of Metal at Core; 300 Diameters

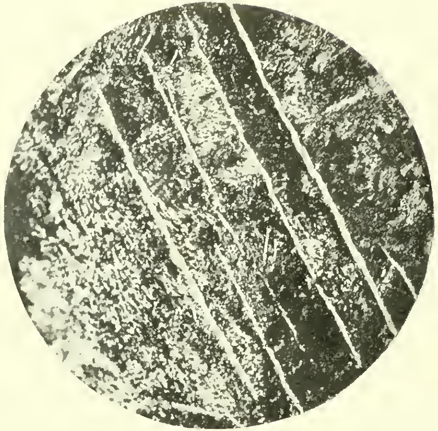

Fig. 21 Open Hearth Steel, 0.20% Carbon, Case Carbonized at 1700 Deg. Fahr. Detail of Outer Edge; 300 Diameters

Note thin laminae of cementite which are most likely to cause spawl in service.

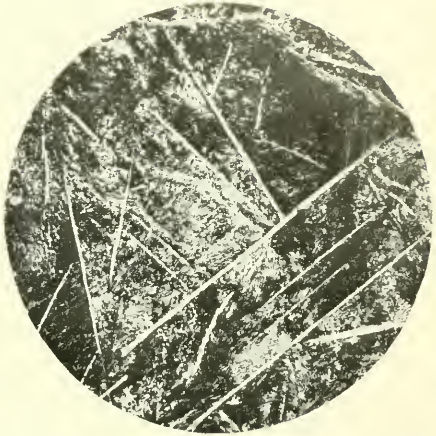

Fig. 22 Open Hearth Steel, 0.20% Carbon, Case Carbonized at 1700 Deg. Fahr. View at Point slightly inside that of Fig. 21; 300 Diameters

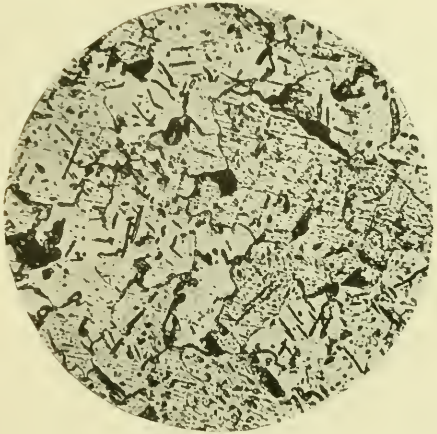

Fig. 23 Open Hearth Steel, 0.20% Carbon, Case Carbonized at 1300 Deg. Fahr. Nitrides at Inner Edge of Gradation Zone; 300 Diameters

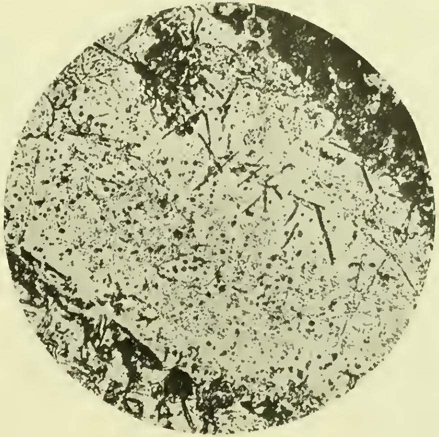

Fig. 24 Open Hearth Steel, 0.20% Carbon, Case Carbonized at 1300 Deg. Fahr. Nitrides in Ferrite Bands shown in Fig. 19; 300 Diameters

Nitrides are black lines seen in white ferrite bands; heavy black mass and black spots are pearlite, details of which could not be shown on account of over printing necessary to bring out nitrides.

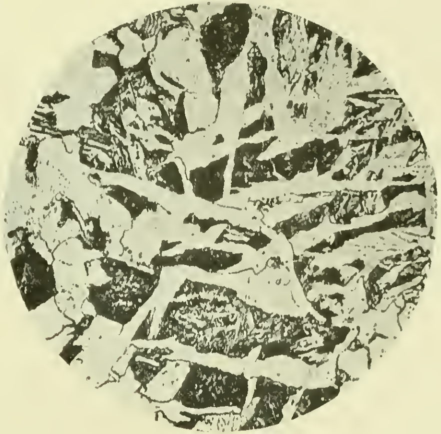

Fig. 25 Nickel Steel, 0.20% Carbon, Case Carbonized at 1700 Deg. Fahr. General Structure of Core; 300 Diameters

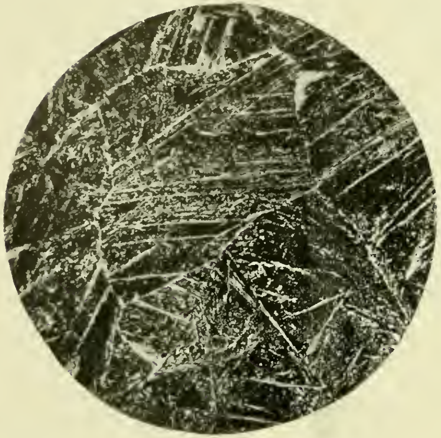

FIG. 26 NICKEL STEEL, 0.20% CARBON, CASE CARBONIZED AT 1700 DEG FAHR. VIEW OF CEMENTITE NEEDLES TAKEN ABOUT IN MIDDLE OF ZONE OF EXCESS CEMENTITE; 300 DIAMETERS

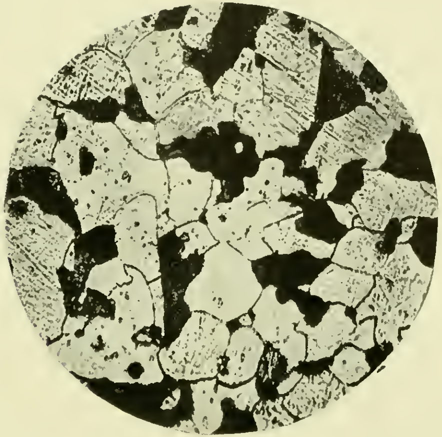

FIG. 27 CHROME VANADIUM STEEL, 0.20% CARBON, CASE CARBONIZED AT 1700 DEG. FAHR. STRUCTURE OF METAL IN CORE; 300 DIAMETERS

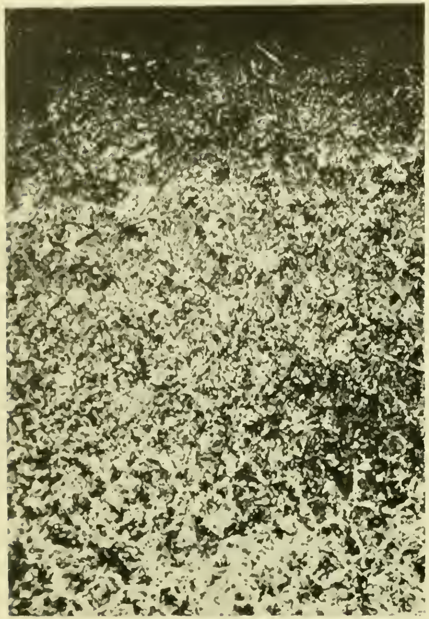

Fig. 28 Nickel Steel, 0.20% Carbon, Case Carbonized at 1700 Deg. Fahr. View shows Deeper Penetration of Carbon in Comparison with Fig. 18; 50 Diameters

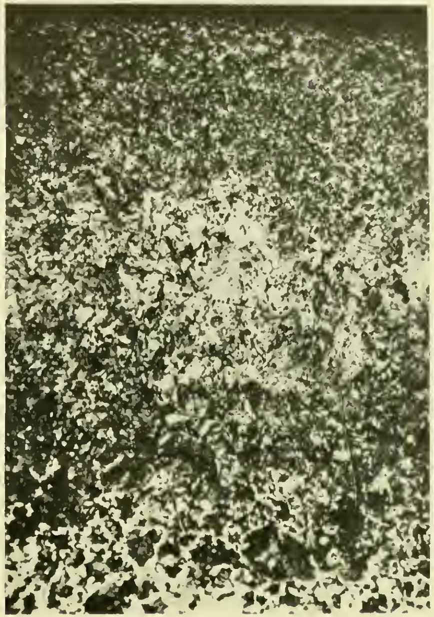

Fig. 29 Chrome Vanadium Steel, 0.20% Carbon, Case Carbonized at 1700 Deg. Fahr. Deep Penetration of Carbon and No Needles of Cementite in Outer Zone in Contrast to Figs. 18 and 28; 50 Diameters

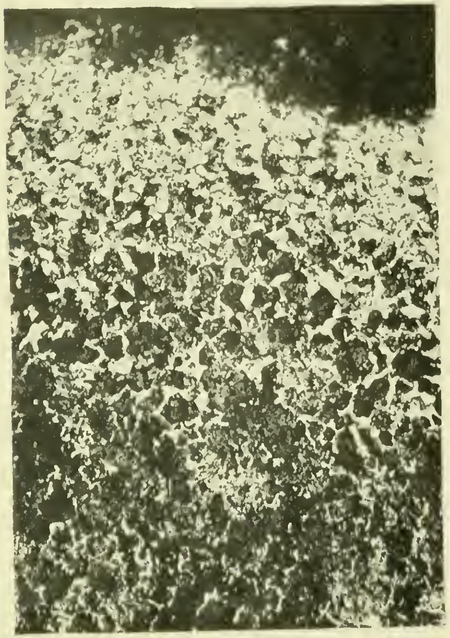

Fig. 30 Chrome Vanadium Steel, 0.20% Carbon, Case Carbonized at 1700 Deg. Fahr. View of Zone of Excess Cementite showing Tendency of Cementite to distribute Itself in Boundaries of Pearlite; 300 Diameters

MICROSCOPIC INVESTIGATION OF OPEN-HEARTH STEELS

TEST PIECE CASE CARBONIZED AT 1300 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 60 SECONDS

63 *Magnification 100 Diameters.* On the outside of this specimen is a zone of pearlite with excess cementite or ferrite in lines too fine for needle test about 0.0075 mm. thick. Inside this is a zone of beautiful lamellar pearlite, about 0.090 mm. thick, of eutectic composition. Further in there is a zone of gradation of carbon content about 0.03375 mm. thick, inside of which is the normal low carbon interior, the pearlite appearing in fine particles surrounded by ferrite. Total penetration of carbon is 0.1350 mm. Just inside the gradation zone and intermingled with it, there is scattered a large number of thin brown needles, varying in size from barely discernible specks to needles about 0.009 mm. long. Mr. K. W. Zimmerschied calls these needles nitrides, since he obtained similar ones by quenching some steel in ammonia. Penetrating the carbonized zone in many places around the circumference are bands of ferrite at various angles from 0 to 90 with the radius containing almost no pearlite, but a thick sprinkling of very minute specks and lines which may be nitrides.

64 *Magnification 400 Diameters.* Under this power the ferrite bands, containing nitrides, appear at intervals all the way around the circumference, more numerous in some parts than in others. These nitrides are more resolved at this power and hence more readily seen, but beyond this increased distinctness, the specimen appears as described above.

TEST PIECE CASE CARBONIZED AT 1400 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 45 SECONDS

65 *Magnification 100 Diameters.* In this specimen the depth of penetration of the carbon seems to be fairly uniform, averaging 0.420 mm. thick. On the outside there is a zone of irregular thickness, containing pearlite with fine lines of excess cementite. The average thickness of this zone is 0.060 mm. Inside the zone of excess cementite comes one of eutectic, about 0.150 mm. thick, followed by a gradation zone 0.225 mm. thick, in which the pearlite exists as large islands surrounded by fine lines of ferrite, growing more numerous toward the interior, until the normal core structure of small islands of pearlite surrounded by ferrite is obtained. The pearlite in the core has gathered into larger islands than in the specimen treated at 1300 deg. fahr. The nitride needles are present inside the eutectic zone, but are not so numerous as in the last specimen. The same is true of the bands of pure ferrite containing nitride needles which penetrate the carbonized zone.

66 *Magnification 400 Diameters.* At this magnification the pearlite throughout the entire carbonized zone is resolved into a beautifully lamellar structure. As before, we have an outer zone of pearlite with excess cementite, showing in places around the circumference small spots of pure cementite on the outer rim. The average thickness of this zone is 0.090 mm. The eutectic zone inside the zone of excess cementite is

approximately 0.120 mm. thick. The gradation zone is 0.1875 mm. thick. giving a total penetration of carbon of 0.3975 mm.

TEST PIECE CASE CARBONIZED AT 1500 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 30 SECONDS

67 *Magnification 100 Diameters.* Under this magnification the effect of carbonizing is noticed in a deeper zone of eutectic, but not in a much greater excess of cementite in the outer portions of the carbonized zone than was observed in the specimen treated at 1400 deg. fahr. On the extreme outer edge of the specimen we see a thin black rim about 0.023 mm. thick, of unresolvable material. Inside this rim we see progressively a zone of pearlite with excess cementite, showing slight crystalline outlines in places, about 0.1275 mm. thick. Next comes the eutectic zone, 0.240 mm. thick, giving a total penetration of carbon of 0.390 mm. This gradation zone varied from large islands of pearlite surrounded by lines of ferrite, to smaller islands of pearlite, surrounded by masses of ferrite. The pearlite on the interior seems to be in larger areas than in the last section surrounded by ferrite. No bands of ferrite, such as supposedly contained nitrides in the last specimens are visible here. Neither are any nitrides visible inside the eutectic zone.

68 *Magnification 400 Diameters.* At this magnification the pearlite in the carbonized zone is resolvable into a beautiful lamellar structure, while that in the core is mostly unresolvable in islands surrounded by ferrite, though some of it appears faintly lamellar. The outer zone of pearlite with excess cementite was 0.150 mm. thick. Inside this was a zone of eutectic 0.150 mm. thick, and then a gradation zone 0.420 mm. thick, giving a total carbon penetration of 0.720 mm. Along the inner part of the gradation zone, we come across a needle or two as we follow the zone around, which might be a nitride, but beyond these few solitary needles no traces, such as the bands cutting into the case, are visible.

TEST PIECE CASE CARBONIZED AT 1600 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 20 SECONDS

69 *Magnification 100 Diameters.* The carbonizing at this temperature appears to have increased the depth of the carbonized zone, but not to have formed a zone of much greater percentage excess cementite than was obtained at the last temperature. The thickness of the outer zone of pearlite and excess cementite varies around the circumference, but averages 0.0975 mm. Inside this zone we find a zone of eutectic averaging 0.345 mm. thick, and a zone of gradation averaging 0.300 mm., giving a total penetration of carbon of 0.7498 mm.

70 *Magnification 400 Diameters.* The pearlite is distinctly lamellar at this magnification, both in the carbonized zone, and in smaller laminations in the interior. The zone of excess cementite varies in thickness, but averages 0.135 mm. Inside this zone comes one of eutectic approximately 0.210 mm. thick, very irregular in depth, and then a gradation zone averaging 0.450 mm., giving a total penetration of carbon of 0.795 mm.

TEST PIECE CASE CARBONIZED AT 1700 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 20 SECONDS

71 *Magnification 100 Diameters.* In this specimen we get both a deeper penetration of carbon than in the last specimen, and a thicker zone of excess cementite. The zone of pearlite with excess cementite was very irregular in depth, but averaged 0.390 mm. In this zone we see a few long cementite needles within the pearlite of varying length and direction, but all of them in straight lines. Inside of this zone comes one of eutectic approximately 0.300 mm. thick, then a gradation zone of about 1.0725 mm., giving a total carbon penetration of 1.7625 mm.

72 *Magnification 400 Diameters.* At this magnification the pearlite appears in large laminations in the carbonized zone and in smaller laminations in the interior. The outer zone, containing pearlite with excess cementite, is approximately 0.225 mm. thick; inside of which comes the eutectic zone, approximately 0.338 mm. thick, and following this a gradation zone 0.60 mm. thick. The outer zone of pearlite with excess cementite contains a large number of cementite needles scattered through it. These needles are all straight lines but point in every direction, the largest of them being about 0.135 mm. long, the widest not over 0.00150 mm. wide. Both the excess cementite and the excess ferrite in their respective zones follow more or less the irregular crystal boundaries.

TEST PIECE CASE CARBONIZED AT 1800 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 15 SECONDS

73 *Magnification 100 Diameters.* In this specimen the excess cementite in the outer zone does not appear so much in the form of crystalline boundaries as it did in the preceding specimen, but more as irregular areas along the outer portion of the zone. The average thickness of this zone is about 0.1950 mm. Inside this zone comes the zone of eutectic. averaging approximately 0.585 mm., followed by a gradation zone averaging 0.975 mm., giving a total penetration of carbon of 1.755 mm. As before, the ferrite in the gradation zone appears in the form of outlines of crystals.

74 *Magnification 400 Diameters.* In this specimen the pearlite is beautifully lamellar in the carbonized zone and lamellar to a slight extent in the interior. On account of the great irregularity in the depth of penetration in the excess cementite in the outer zone, it is difficult at this magnification to obtain an average depth. The quantity of cementite needles within the pearlite grains has increased. The following figures are approximate: For the zone of excess cementite the depth is 0.293 mm. The depth of the eutectic zone is 0.450 mm. and that of the gradation zone 0.90 mm., giving a total penetration of carbon of 1.64 mm.

MICROSCOPIC INVESTIGATION OF CASE-CARBONIZED NICKEL STEELS

TEST PIECE CASE CARBONIZED AT 1300 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 40 SECONDS

75 *Magnification 100 Diameters.* At this magnification this specimen shows only a slight carbonization on the outside, apparently not even reaching the eutectic. In places around the circumference bands of darker material, looking like eutectic, extend quite a little distance into the section, very similar in appearance to the bands containing nitrides noticed in the open-hearth series. These bands occur only in two or three places. The interior of the section consists of finely divided pearlite with excess ferrite. Since the effect of carbonization is slight, it is difficult to tell where the carbonized zone ends and the normal interior structure begins, but an approximate figure of 0.045 mm. thickness gives a general idea of the depth of penetration of the carbon.

76 *Magnification 400 Diameters.* On the flat outer rim of this section appears to be a little pure ferrite, but this is a very thin rim, so that a needle test does not distinguish it from cementite. The pearlite in the carbonized zone is not resolvable at this power. The average depth of the carbonized zone is 0.180 mm. and no eutectic is present.

TEST PIECE CASE CARBONIZED AT 1400 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 40 SECONDS

77 *Magnification 100 Diameters.* The outer zone at this magnification appears to be of eutectic composition and an average thickness of 0.195 mm., inside of which comes a gradation zone approximately 0.390 mm. thick. The pearlite in the interior has gathered together slightly more than in the last specimen, and as before, is surrounded by ferrite. In this specimen we see ferrite bands at different places around the circumference penetrating the outer zone, which look almost the same as those found in the open-hearth series, especially in the specimen treated at this temperature, which contains nitrides.

78 *Magnification 400 Diameters.* The pearlite in the carbonized zone is faintly resolvable into the lamellar condition at this magnification. The highest carbon appears to be eutectic in a zone at the outside of the piece approximately 0.203 mm. thick. In one or two places there appear to be a slight excess of ferrite on the very outside, but this may be due to the rounding of the edges. The bands already mentioned, which penetrated the carbonized zone, are ferrite containing either nitrides or pearlite. At this power no needles which definitely resemble the nitrides are visible.

TEST PIECE CASE CARBONIZED AT 1500 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 20 SECONDS

79 *Magnification 100 Diameters.* In this specimen we get an irregular carbonized zone apparently not going beyond the eutectic in carbon content on the outside. The zone of eutectic, however, is much thicker than in the last specimen. The bands, which penetrated the carbonized zone

in the last specimen are here present in about the same number. The approximate thickness of the eutectic zone is 0.338 mm. The gradation zone is 0.585 mm., giving a total carbon penetration of 0.923 mm.

80 *Magnification 400 Diameters.* At this power we see an outer zone of eutectic approximately 0.278 mm. and a gradation zone of 0.510 mm. The total penetration of carbon is approximately 0.788 mm. The pearlite on the interior is just discernible as lamellar at this magnification and that in the carbonized zone is a little more easily recognized and also lamellar. The bands penetrating the eutectic on the outside appear to consist of pure ferrite with dark dots and smaller irregular lines scattered throughout.

TEST PIECE CASE CARBONIZED AT 1600 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 15 SECONDS

81 *Magnification 100 Diameters.* At this magnification there is little, if any, excess cementite in the outer zone of carbonization. There is, however, a thick zone of eutectic and inside this a zone of decreasing carbon. Also, in one or two places around the circumference we see the penetration bands which have been discussed in the last two or three specimens. The average thickness of the eutectic is 0.390 mm. That of the gradation zone is 0.683 mm. The total penetration of carbon is 1.073 mm.

82 *Magnification 400 Diameters.* At this magnification there appears to be a little excess cementite on the outer edge of the eutectic zone in places. This cementite appears in very fine lines like outlines of ferrite crystals. Since the cementite penetrates to varying depths and is present only in a small amount, we will consider the zone of eutectic as running out to the edge of the specimen and give for an average thickness of eutectic 0.338 mm., inside of which comes a gradation zone approximately 0.563 mm. The total penetration of carbon is 0.901 mm. The pearlite in the carbonized zone is resolvable in places into laminations. The maximum penetration of the excess cementite is about 0.180 mm.

TEST PIECE CASE CARBONIZED AT 1700 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 15 SECONDS

83 *Magnification 100 Diameters.* In this specimen the outer zone, containing pearlite with excess cementite, shows the needles of cementite scattered through it pretty freely, and as lines in the form of crystalline division. The whole outer zone, with the exception of the cementite needles and a few small white spots of cementite, appears to be pearlite. The thickness of this zone is 0.760 mm. Inside of this comes a gradation zone approximately 0.975 mm. The total penetration of carbon is 1.755 mm. No bands such as were seen in the last specimens and which penetrated the carbonized zone appear here, except in one spot where there is a slight penetration, the material looking like cementite.

84 *Magnification 400 Diameters.* At this power the zone of excess cementite containing needles of cementite is readily distinguished, and though varying in depth is approximately 0.473 mm. thick, inside of which

comes a zone of eutectic about 0.225 mm. thick. Inside of this is a gradation zone 0.675 mm. thick. The total penetration of carbon is 1.373 mm. The two bands of cementite already mentioned look like those described under the last specimen and may contain either this peculiar form of pearlite or nitrides.

TEST PIECE CASE CARBONIZED AT 1800 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 15 SECONDS

85 *Magnification 100 Diameters.* This specimen shows a greater penetration of the excess cementite than the last one, but the cementite appears in the form of irregular globules rather than as crystalline boundaries, in which form it appeared in the open-hearth series. The average depth of the zone of excess cementite is 0.293 mm. inside of which is the eutectic zone approximately 0.488 mm. thick, and inside of which a zone of decreasing carbon content approximately 1.28 mm. The total penetration of carbon is 2.061 mm. The bands penetrating the crystalline zone which appear in the last specimens are not present.

86 *Magnification 400 Diameters.* At this magnification the excess cementite in the outer zone is easily seen and appears principally as globules and finer lines, though only a few needles are present. The thickness of this zone is approximately 0.450 mm., inside of which comes a zone of eutectic approximately 0.428 mm. followed by a zone of decreasing carbon 0.15 mm., giving a total carbon penetration of 2.002 mm. The pearlite in the carbonized zone is only slightly resolvable at this magnification into lamellar pearlite. That on the interior is not resolvable at this power and may be troostitic.

MICROSCOPIC INVESTIGATION OF CHROME VANADIUM STEEL

TEST PIECE CASE CARBONIZED AT 1300 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 45 SECONDS

87 *Magnification 100 Diameters.* On the outside of this specimen is a black etching zone, unresolvable at this power, but probably eutectic pearlite, about 0.03 mm. thick, inside of which is a gradation zone 0.075 mm. thick. Inside these two zones is a black etching zone approximately 0.0675 mm. thick, which appears to be pearlite gathered together more or less in areas surrounded by lines of ferrite. The thicknesses of these individual zones varied but the carbon penetration was pretty and averaged 0.1725 mm.

88 *Magnification 400 Diameters.* On the outside there is a zone of pearlite containing small white globules that are probably cementite, though they are too small to apply the needle test, to distinguish them from ferrite. In places there is a very thin outer rim of cementite which confirms the above statement regarding the white globules. Inside this is a zone of very dark etching, faintly resolvable lamellar eutectic pearlite, approximately 0.03 mm. thick, inside of which is a gradation zone about 0.01575 mm. thick. This zone is difficult to measure since the carbon content grades off very slowly.

TEST PIECE CASE CARBONIZED AT 1400 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 30 SECONDS

89 *Magnification 100 Diameters.* On the outside there is a thick zone of eutectic showing possibly a slight excess cementite in fine lines on the outer edge in some places, the average thickness of the zone being about 0.15 mm. Inside of this zone is a gradation zone approximately 0.2475 mm. thick. As in the last specimen the decrease in carbon content is very gradual, making this zone difficult to measure. Total penetration of carbon is 0.03975 mm.

90 *Magnification 400 Diameters.* Here we see on the outside a thin rim about 0.0090 mm. thick of pearlite in excess cementite, inside of which comes a eutectic zone about 0.135 mm. thick, and then a gradation zone about 0.300 mm. thick, giving a carbon penetration of 0.444 mm.

TEST PIECE CASE CARBONIZED AT 1500 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 20 SECONDS

91 *Magnification 100 Diameters.* This specimen shows on the outside a definite zone of pearlite with excess cementite about 0.0525 mm. thick, inside of which is a eutectic zone about 0.4875 mm. and then a gradation zone of 0.195 mm. thick. Total penetration of carbon is 0.735 mm.

92 *Magnification 400 Diameters.* At this magnification there is a zone of pearlite with large excess cementite. This zone is about 0.06 mm. thick. inside it is a zone of pearlite, with just a little excess cementite, in small polygonal outlines, the average thickness of the zone being about 0.225 mm. thick, and then a eutectic zone about 0.180 mm. thick, followed by a gradation zone approximately 0.3375 mm. thick. The pearlite is not resolvable either in the carbonized zone or in the core. Total penetration of carbon is 0.8025 mm.

TEST PIECE CASE CARBONIZED AT 1600 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 150 SECONDS

93 *Magnification 100 Diameters.* Here we see a zone of pearlite with a large excess of cementite about 0.0375 mm., inside of which is a zone of eutectic about 0.51 mm. thick, and then a gradation zone 0.248 mm. thick. Total penetration of carbon is 0.795 mm.

94 *Magnification 400 Diameters.* With this magnification we see an outer zone about 0.045 mm. thick, consisting of cementite with pearlite in small areas throughout it. Then we come to a zone of pearlite containing excess cementite in fine polygonal outlines, this zone being about 0.1125 mm. Inside this zone is a eutectic zone approximately 0.2475 mm. thick, followed by a gradation zone approximately 0.2775 mm. thick, giving a total penetration of carbon of about 0.6825 mm. The pearlite in the carbonized zone is mostly unresolvable, though some of it shows very fine laminations. That on the interior is resolvable into fine lamellar pearlite.

TEST PIECE CASE CARBONIZED AT 1700 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 15 SECONDS

95 *Magnification 100 Diameters.* On the outside of this specimen we see a sharply defined zone of pearlite with excess cementite approximately

0.067 mm. thick. Next comes a zone of eutectic about 0.8755 mm. thick, followed by a gradation zone about 0.4875 mm. thick. Total penetration of carbon is 1.430 mm.

96 *Magnification 400 Diameters.* Here, as in the last specimen, we get a sharply defined zone of cementite with small areas of pearlite scattered through it, apparently a little more pearlite than half and half, this zone being about 0.105 mm. thick. Inside this is a zone about 0.360 mm. thick, which contains pearlite surrounded by thin lines of cementite in polygonal outlines. Then comes the eutectic zone about 0.405 mm. thick, followed by a zone of gradation approximately 0.45 mm. thick. The carbon content in the latter changes very slowly so that the zone is difficult to measure. Total penetration of carbon is 1.320 mm.

TEST PIECE CASE CARBONIZED AT 1800 DEG. FAHR., ETCHED IN 5 PER CENT PICRIC ACID FOR 15 SECONDS

97 *Magnification 100 Diameters.* This specimen shows a zone on the ourside of pearlite surrounded by fine lines of cementite, this zone being about 0.15 mm. thick. There comes a zone of eutectic approximately 0.975 mm. thick, followed by a gradation zone approximately 0.795 mm. thick, giving a total carbon penetration of 1.92 mm. In various places throughout the carbonized zone appear irregularly shaped areas not over 0.675 mm. approximately, or their longest dimension, which are unresolvable at this power.

98 *Magnification 400 Diameters.* At this magnification, we see a wide zone, about 0.3375 mm. thick, containing pearlite with excess cementite in fine lines showing polygonal figures, like boundaries between ferrite crystals. Inside this comes a zone of eutectic 0.5625 thick, followed by a gradation zone 0.675 mm. thick, giving a total penetration of carbon of 1.575 mm. The white spots above noticed were shown to be cementite by scratching with a needle. In some places there are little white spots containing fine brown lines possibly looking like lamellar pearlite, in which the ferrite layer is very thin, much thinner than the cementite, in some cases only one-seventh to one-tenth as wide. This structure has been noticed in nearly all the areas containing excess cementite.

APPENDIX NO. 3

EFFICIENCY OF CASE-CARBONIZING MATERIALS

CASE-CARBONIZING MATERIAL NO. 1

Etched in 2.5 Per Cent Picric Acid for 45 Minutes

99 *Run No. 1, Bar No. 1.* On the outside of this specimen is a uniform zone 0.455 mm. thick of pearlite containing lines of excess cementite; inside is a zone of eutectic pearlite 0.455 mm. thick, followed by a gradation zone 0.650 mm. thick, inside of which comes the normal structure of about 0.20 per cent carbon steel consisting of ferrite containing small islands of pearlite. The pearlite throughout the section is lamellar. Total penetration of carbon is 1.560 mm.

100 *Run No. 2, Bar No. 7.* This specimen shows the outer zone of eutectic pearlite 0.715 mm. thick, inside of which comes a gradation zone 1.001 mm. thick, followed by the normal interior of ferrite containing islands of pearlite. The pearlite throughout the section is lamellar. Total penetration of carbon is 1.716 mm.

Etched in 2.5 Per Cent Picric Acid for 30 Seconds

101 *Run No. 3, Bar No. 13.* This specimen shows an outer zone of eutectic 0.358 mm. thick, inside of which is a gradation zone 1.138 mm. thick, surrounding the normal low carbon interior structure. Total penetration of carbon is 1.496 mm.

Etched in 2.5 Per Cent Picric Acid for 60 Seconds

102 *Run No. 1, Bar No. 2.* This specimen shows an outer zone of lamellar pearlite containing lines of excess cementite, the zone being 0.450 mm. thick. Inside this zone comes a zone of eutectic 0.650 mm. thick, followed by a gradation zone 0.748 mm. thick, giving a total penetration of carbon of 1.848 mm. Inside these zones is the normal low carbon structure.

103 *Run No. 2, Bar No. 8.* On the outside of this specimen is a zone of eutectic 0.650 mm. thick, inside of which comes a gradation zone 1.073 mm. giving a total penetration of carbon of 1.723 mm. The outer zone contains numerous small white areas, probably of ferrite, but too small to scratch with a needle.

Etched in 2.5 Per Cent Picric Acid for 70 Seconds

104 *Run No. 3, Bar No. 14.* In this specimen the ferrite penetrates to the very edge in many places around the circumference and the effect of the carbonizing has been simply to raise the carbon content of the outer portion, forming a zone of decreasing carbon about 0.975 mm. thick.

CASE-CARBONIZING MATERIAL NO. 3

Etched in 2.5 Per Cent Picric Acid for 45 Seconds

105 *Run No. 1, Bar No. 3.* Outer zone of eutectic is 0.520 mm. thick, inside this is a zone of gradation 1.235 mm. thick, giving a total penetration of carbon of 1.755 mm. The outer zone contains small white spots looking like ferrite, but too small to scratch.

106 *Run No. 2, Bar No. 9.* On the outside a rather irregular zone of eutectic approximately 0.520 mm. thick, inside of which is a gradation zone 1.300 mm. giving a total penetration of carbon of 1.820 mm. thick. The outer zone of eutectic contains here and there small islands probably of ferrite, too small to scratch with the needle.

Etched in 2.5 Per Cent Picric Acid for 60 Seconds

107 *Run No. 3, Bar No. 15.* Outer zone of eutectic in places, but penetrated quite often along the circumference by ferrite bands so that it is questionable whether even the very outside has reached the carbon content of the eutectic. From the outside the carbon content decreases to that of the normal low carbon interior, this gradation zone being 0.325 mm. thick, which is the total penetration of carbon.

CASE-CARBONIZING MATERIAL NO. 1

Etched in 2.5 Per Cent Picric Acid for 60 Seconds

108 *Run No. 1, Bar No. 4.* On the outside of this specimen is a zone of eutectic 0.975 mm. thick, inside of which is a gradation zone 1.300 mm. thick, giving a total penetration of carbon of 2.275 mm. In places along the outer zone there are small islands of white material, either cementite or ferrite too small to scratch with the needle and containing very fine lines of pearlite in lamellation.

Etched in 2.5 Per Cent Picric Acid for 50 Seconds

109 *Run No. 2, Bar No. 10.* On the outside is a zone of eutectic approximately 0.878 mm. thick, inside of which comes a zone of gradation 0.975 mm. thick, giving a total penetration of 1.853 mm. As in the other specimens there are small spots of white material scattered through the outer zone.

110 *Run No. 3, Bar No. 16.* On the outside is a zone of eutectic approximately 0.455 mm. thick, quite irregular and in places having ferrite penetrating it to the edge of the specimen. Inside of this zone comes one of decreasing carbon 0.975 mm. thick, giving a total penetration of carbon of 1.43 mm.

CASE-CARBONIZING MATERIAL NO. 5

Etched in 2.5 Per Cent Picric Acid for 60 Seconds

111 *Run No. 1, Bar No. 5.* On the outside of this specimen is a zone of lamellar pearlite containing lines of excess cementite in polygonal outline as usual, this zone being about 0.780 mm. thick; inside comes the eutectric zone 0.520 mm. thick, followed by a zone of gradation 0.975 mm. thick, giving a total penetration of carbon of 2.275 mm.

112 *Run No. 2, Bar No. 11.* On the outside is a eutectic zone approximately 0.780 mm. thick but quite irregular in depth. Then comes a gradation zone 0.975 mm. thick, giving a total penetration of carbon of 1.755 mm. As usual in the outside zone are small white spots of ferrite or cementite too small to scratch with the needle.

113 *Run No. 3 Bar No. 17.* In this specimen is an outer zone of eutectic 0.225 mm. thick, containing several white spots of ferrite, inside of which comes a gradation zone of 1.300 mm. thick, giving a total penetration of carbon of 1.525 mm.

CASE-CARBONIZING MATERIAL NO. 6

Etched in 2.5 Per Cent Picric Acid for 55 Seconds

114 *Run No. 1, Bar No. 6.* This carbonizing compound did not even bring the outer skin of the metal up to eutectic but carbonized the outside slightly to a depth of 1.235 mm., which is the total penetration of carbon.

Etched in 2.5 Per Cent Picric Acid for 60 Seconds

115 *Run No. 2, Bar No. 12.* This specimen shows an outer zone of increased carbon content not as high as eutectic, the zone being 0.975 mm. thick, which is the total penetration of carbon.

116 *Run No. 3, Bar No. 18.* On the outside of this specimen is a zone of increased carbon content not as high as eutectic, 0.040 mm. thick.

CASE-CARBONIZING MATERIAL NO. 7

Etched in 2.5 Per Cent Picric Acid for 55 Seconds

117 *Run No. 1, Bar No. 19.* On the outside of this specimen is a zone of lamellar pearlite containing lines of excess cementite more or less in polygons, the depth of the zone being 0.325 mm. thick, inside this comes a eutectic zone 0.520 mm. thick, and inside this a gradation zone 0.975 mm. thick, giving a total penetration of carbon of 1.920 mm.

Etched in 2.5 Per Cent Picric Acid for 50 Seconds

118 *Run No. 2, Bar No. 25.* On the outside is a zone of eutectic approximately 2.60 mm. thick, containing small spots of white material probably ferrite in one or two places; inside comes a gradation zone 1.300 mm. thick, giving a total penetration of carbon of 1.560 mm.

Etched in 2.5 Per Cent Picric Acid for 60 Seconds

119 *Run No. 3, Bar No. 31.* On the outside is a zone of eutectic approximately 0.715 mm. thick, containing a little excess ferrite in spots around it, in some places the ferrite extending almost to the edge. Inside of this comes a gradation zone 1.170 mm. thick, giving a total penetration of carbon of 1.995 mm.

CASE-CARBONIZING MATERIAL NO. 8

Etched in 2.5 Per Cent Picric Acid for 50 Seconds

120 *Run No. 1, Bar No. 20.* This specimen shows an outer zone of lamellar pearlite containing polygonal lines of excess cementite, the zone

being approximately 0.455 mm. thick; inside comes a eutectic zone 0.520 mm. thick and a gradation zone 0.975 mm. thick, giving a total penetration of carbon of 1.950 mm.

121 *Run No. 2, Bar No. 26.* On the outside is a zone of lamellar pearlite containing fine lines of excess cementite following the boundaries of the ferrite crystals as usual, the thickness of the zone averaging 0.325 mm.; inside is a zone of eutectic 0.715 mm. thick, followed by a gradation zone 0.910 mm. thick, giving a total penetration of carbon of 1.950 mm.

Etched in 2.5 Per Cent Picric Acid for 55 Seconds

122 *Run No. 3, Bar No. 32.* On the outside is a zone of lamellar pearlite containing a little excess cementite in very fine lines following the boundaries of the ferrite crystals. The thickness of this zone is 0.195 mm. Inside of this comes a eutectic zone 0.650 mm. thick, followed by a gradation zone 0.650 mm. thick, giving a total penetration of carbon of 1.495 mm.

CASE-CARBONIZING MATERIAL NO. 9

Etched in 2.5 Per Cent Picric Acid for 45 Seconds

123 *Run No. 1, Bar No. 21.* On the outside of this specimen is a zone 1.235 mm. thick of lamellar pearlite containing lines of excess cementite in rather thick lines around the ferrite crystals and showing small cementite needles in places, which appear to branch off from the main crystaline boundaries. Inside comes a eutectic zone 0.423 mm. thick, followed by a gradation zone 0.780 mm. thick, giving a total penetration of carbon of 2.438 mm.

Etched in 2.5 Per Cent Picric Acid for 55 Seconds

124 *Run No. 2, Bar No. 27.* On the outside is a zone 1.075 mm. thick of lamellar pearlite containing lots of excess cementite in polygonal outlines, but no cementite needles such as were noticed in the last specimen. Inside this comes a eutectic zone 0.455 mm. thick, followed by a gradation zone 0.780 mm. thick, giving a total penetration of carbon of 2.300 mm.

Etched in 2.5 Per Cent Picric Acid for 50 Seconds

125 *Run No. 3, Bar No. 33.* On the outside is a zone of lamellar pearlite containing lines of excess cementite outlining the ferrite crystals as usual, the depth of the zone being 0.325 mm. Inside this comes a zone of eutectic 0.845 mm., followed by a gradation zone 0.975 mm. thick, giving a total penetration of carbon of 2.145 mm.

CASE-CARBONIZING MATERIAL NO. 10

Etched in 2.5 Per Cent Picric Acid for 55 Seconds

126 *Run No. 1, Bar No. 22.* On the outside of this specimen is a zone of eutectic approximately 0.975 mm. thick, containing spots here and there of white material probably ferrite, such as were noticed in the other specimens. Inside comes a gradation zone 1.105 mm. thick, giving a total penetration of carbon of 2.080 mm.

Etched in 2.5 Per Cent Picric Acid for 50 Seconds

127 *Run No. 2, Bar No. 28.* On the outside is a eutectic zone rather irregular in depth but averaging 0.065 mm., so that the ferrite penetrates to the circumference in some places. Inside comes one of decreasing carbon 1,170 mm. thick, giving a total penetration of carbon of 1.235 mm.

Etched in 2.5 Per Cent Picric Acid for 15 Seconds

128 *Run No. 3, Bar No. 34.* On the outside is a rim of pure ferrite 0.065 mm., inside of which is a carbonized zone in which the carbon content decreases to that of the interior. This zone is 0.910 mm. thick, giving a total penetration of carbon of 0.975.

CASE-CARBONIZING MATERIAL NO. 11

Etched in 2.5 Per Cent Picric Acid for 55 Seconds

129 *Run No. 1, Bar No. 23.* On the outside of this specimen is a zone of eutectic varying in depth, in some places almost reaching the edge, but of an average depth of 0.195 mm. Inside of this comes a gradation zone 1.300 mm. thick, giving a total penetration of carbon of 1.495 mm.

130 *Run No. 2, Bar No. 29.* This specimen was not carbonized up to the eutectic even on the outside, so that there is simply a zone of decreasing carbon content approximately 1.658 mm., which is the total penetration of carbon.

Etched in 2.5 Per Cent Picric Acid for 50 Seconds

131 *Run No. 3, Bar No. 35.* On the outside is a thin rim, maximum thickness 0.007 mm., of ferrite not continuous, but present on most of the circumference. Inside comes a zone of gradation 1.235 mm. thick, giving a total penetration of carbon of 1.242 mm.

CASE-CARBONIZING MATERIAL NO. 12

Etched in 2.5 Per Cent Picric Acid for 60 Seconds

132 *Run No. 1, Bar No. 24.* On the outside of this specimen is a rather irregular zone 0.329 mm. thick, of eutectic containing white specks here and there as noted in the other specimens. Inside comes one of gradation 1.235 mm. thick, giving a total penetration of carbon of 1.574 mm.

Etched in 2.5 Per Cent Picric Acid for 65 Seconds

133 *Run No. 2, Bar No. 30.* On the outside of this specimen we see a zone of eutectic very irregular and penetrated now and again by ferrite bands. The average thickness of this zone is 0.130 mm. Inside of this comes a gradation zone 1.560 mm. thick, giving a total penetration of carbon of 1.690 mm.

134 *Run No. 3, Bar No. 36.* On the outside is a rim of pure ferrite in some places, in others this is not present. The maximum thickness is 0.007 mm., and inside this comes a gradation zone 1.560 mm. thick, totaling 1.567 mm.

CASE-CARBONIZING MATERIAL NO. 13

Etched in 2.5 Per Cent Picric Acid for 55 Seconds

135 *Run No. 1, Bar No. 37.* On the outside of this specimen is a zone of lamellar pearlite containing excess cementite in thin polygonal outlines. The depth of penetration of these outlines varies, but averages 0.325 mm. Inside comes a zone of eutectic 0.520 mm. thick, followed by a gradation zone 1.300 mm., giving a total penetration of carbon of 2.145 mm.

Etched in 2.5 Per Cent Picric Acid for 70 Seconds

136 *Run No. 2, Bar No. 39.* On the outside is a zone of eutectic approximately 0.520 mm. thick, but very irregular, in some cases the ferrite globules penetrate almost to the edge. Inside comes a gradation zone of 1.170 mm., giving a total penetration of carbon of 1.690 mm.

Etched in 2.5 Per Cent Picric Acid for 80 Seconds

137 *Run No. 3, Bar No. 41.* On the outside is a zone of eutectic averaging 0.291 mm., but very irregular and containing spots of ferrite here and there, penetrated more or less by the ferrite bands from the interior. Inside comes a gradation zone of 1.170 mm. thick, giving a total penetration of carbon of 1.463 mm.

CASE-CARBONIZING MATERIAL NO. 14

Etched in 2.5 Per Cent Picric Acid for 55 Seconds

138 *Run No. 1, Bar No. 38.* On the outside of this specimen is a zone of lamellar pearlite containing a great deal of excess cementite, both as polygonal outlines and in a few places as long thick cementite needles. The depth of penetration of this excess cementite varies, but averages 1.040 mm. Inside comes a eutectic zone 0.339 mm. thick, followed by a gradation zone 0.650 mm. thick, giving a total penetration of carbon of 2.029 mm.

139 *Run No. 2, Bar No. 40.* On the outside is an irregular zone of lamellar pearlite containing excess cementite, both as polygonal outlines and large needles, and averaging 1.300 mm. thick. Inside comes a zone of eutectic 0.450 mm. thick, followed by a gradation zone 1.105 mm., giving a total penetration of carbon of 2.855 mm.

140 *Run No. 3, Bar No. 42.* Outer zone of lamellar pearlite containing excess cementite averaging 1.170 mm, both as polygonal outlines and as needles. Inside comes a zone of eutectic of 0.845 mm. thick, followed by a gradation zone of 1.300 mm. giving a total penetration of carbon of 3.315 mm.

FOREIGN REVIEW

BRIEF ABSTRACTS OF CURRENT ARTICLES IN FOREIGN PERIODICALS

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FOREIGN REVIEW

It is the purpose to cover in the Foreign Review as wide a range of subjects as space permits, endeavoring at the same time to present the salient points of the articles abstracted and the most valuable engineering data contained in them. The usefulness of the Review will be further increased through arrangements lately made by the library of the Engineering Societies, including that of The American Society of Mechanical Engineers, to add to the number of publications in the Slavonic languages received at the library. For the mechanical engineer this is of importance owing to the wide application of the oil engine in Russia which is in this respect in a position very similar to that of the United States.

THIS MONTH'S ARTICLES

Among this month's articles will be found one on the thermodynamics of the gas turbine by Dejmek abstracted from the journal of the Bohemian Society of Engineers and Architects and one by the designer of the first large gas turbine in actual operation, engineer Holzwarth. With respect to reciprocating internal combustion engines attention is called to the description of the Sabathé engine and the accompanying discussion of the mixed combustion process, as well as to the article by Mérigeault on the influence of the speed of combustion on the efficiency of a gas engine. Mader's article on the micro-indicator describes an apparatus intended to be used with high speed small internal combustion engines where an ordinary indicator does not give satisfactory results.

The experiments of Christlein reported in The Journal for February 1912, by showing how comparatively inexact has been our knowledge of the processes of flow of elastic fluids through nozzles, have attracted a good deal of attention to this subject. An article investigating the relation between the velocity of flow and the so-called acoustic velocity by Zerkowitz is to be found in abstract in this month's Review. K. Schultze describes a method for determining the steam consumption of a turbine plant without the use of expensive steam meters or the application of methods of measurement beyond the ability of the usual plant attendants. Attention is also called to the article on the application of compressed air in the atomization of liquid fuel in locomotives, mainly because the author claims to have obtained an unusual economy of 38 per cent where American engineers have practically found no improvement at all.

A series of articles on the Cardan drive, ball bearings, lubrication tests, etc. will be found in the section on Mechanics, while in the section of Air Machinery are described several interesting types of air conditioning apparatus of German design. Of articles on autogenous welding one gives results of an investigation of constructional requirement of an oxy-acetylene welding burner and the other data of an important investigation on the strength of welded seams by Desjuzeur, director of the Lyons association of Owners of Steam Machinery, whose investigation on boiler sheets has been reported in the preceding number.

A new process for the manufacture of iron and steel is also reported, and attention is called in this connection to the numerous attempts of applying coal dust as fuel in internal combustion engines made in England lately. Data on the efficiency of sand blasting machines are also given. In conclusion may be mentioned the report of the work of Imbert on measuring the labor output of a workman, and graphically showing the relative efficiency of a skilled workman and an apprentice.

The Editor will be pleased to receive inquiries for further information in connection with articles reported in this Review. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *t* theoretical. Articles of exceptional merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

Aeronautics

STARTING COURSE (*Anlaufweg*, G. König. *Zeits. für Flugtechnik und Motorluftschiffahrt*, vol. 3, no. 19, p. 247. October 12, 1912. 1 p., 1 fig. *t*). A mathematical demonstration that the length of the start run is proportional directly to specific load on the planes and normal velocity of flow, and inversely to the specific efficiency (referred to unit weight of the flying machine) in the equation for length of start, there being also a factor to take care of the constructional pecularities of each type. The length of the start run increases in squares when the net weight of the airship is increased by some useful load (in the first degree).

APPARATUS FOR MEASURING STRESSES IN THE STAYWIRES OF AIRSHIPS (*Apparat zur Messung der in den Drahten eines Flugapparates auftretenden Spannungen*, Wittmaack. *Zeits. für Flugtechnik und Motorluftschiffahrt*, vol. 3, no. 19, p. 250. October 12, 1912, 2 pp., 2 figs. *d*). Description of the Largier apparatus described in The Journal, May 1912, p. 806. The author of the present article states that the apparatus has not yet been tried for measuring stresses during actual flight.

Air Machinery and Ventilation

UNDERGROUND TRACTION BY COMPRESSED AIR LOCOMOTIVES IN FRENCH MINES (*La traction souterraine par locomotives à air comprimé dans les mines françaises*, E. Leroux. *Bulletin de la Société de l'Industrie Minérale*, September 1912. p. 299. 36 pp., 22 figs. *de*). Description of some compressed air mine locomotives and data of tests, among others those of the Leroux locomotive.

PNEUMATIC POST IN ITALY (*La posta pneumatica in Italia. L'Industria*, vol. 26, no. 39, p. 613. September 29, 1912. 7 pp., 26 figs. *d*). Description of the pneumatic post installation in Milan, Italy. German apparatus of the closed-circuit type is used, the parcels in the tubes moving at a speed from 500 m (1640 ft. contract speed) to 700 m (2296 ft. speed at tests). Each parcel may contain from 15 to 20 letters and telegrams.

PRESSURE PIPING OF AN UNDERGROUND MINE PUMPING ENGINE (*Druckleitung einer unterirdischen Wasserhaltungsmaschine*, Ed. Kaschny. *Der praktische Maschinen-Konstrukeur*, vol. 45, no. 21, p. 350. October 10, 1912. 2 pp., 3 figs. and one sheet of drawings. *d*). *Wrought-iron or seamless steel tubes* have the following advantages as compared with *cast-iron ones*: greater length and therefore fewer flange connections per unit of length; thinner walls, and therefore easier handling, lower stress of the foundations and lower expenses of installation. If cast-iron pipes are still used, however, this may be partly explained by the usual unwillingness of mine owners to try new things, and partly by the idea that cast iron is better able to resist corrosion due to the action of mine waters; as to the latter it has been found, however, that a properly applied coat of tar affords a practically perfect protection to wrought-iron and steel pipes. The rest of the article contains a detailed description of the piping in a Russian mine the name of which is not given.

NEW VENTILATING APPARATUS IN 1911 (*Neuerungen in der Lüftungstechnik aus dem Jahre 1911. Zeits. für Beleuchtungswesen*, vol. 18, no. 28, p. 327. October 10, 1912. 2 pp., 6 figs. *d*). Descriptions of new types of *air conditioning apparatus* of German design. The author states that it was in the United States that the tendency to build apparatus which would simultaneously wash the air and condition it originated. The device shown in Fig. 1A, was patented in Germany (patent No. 239583) by H. Assmuth, a resident of Daressalam, German East Africa, and is primarily designed for use in *tropical countries*. It combines an air cooling apparatus with an air distributor, the cooling being based on the variation of the temperature of the air with changes in its pressure. P is an air pump surrounded by the jacket W; B is a tank containing the suction pipe serpentine RC_1 and pressure pipe serpentine RA, the first connecting with the air reservoir L, also surrounded by a jacket W; V is the distributor, and R air cleaner. The jackets W are filled with water, the tank B

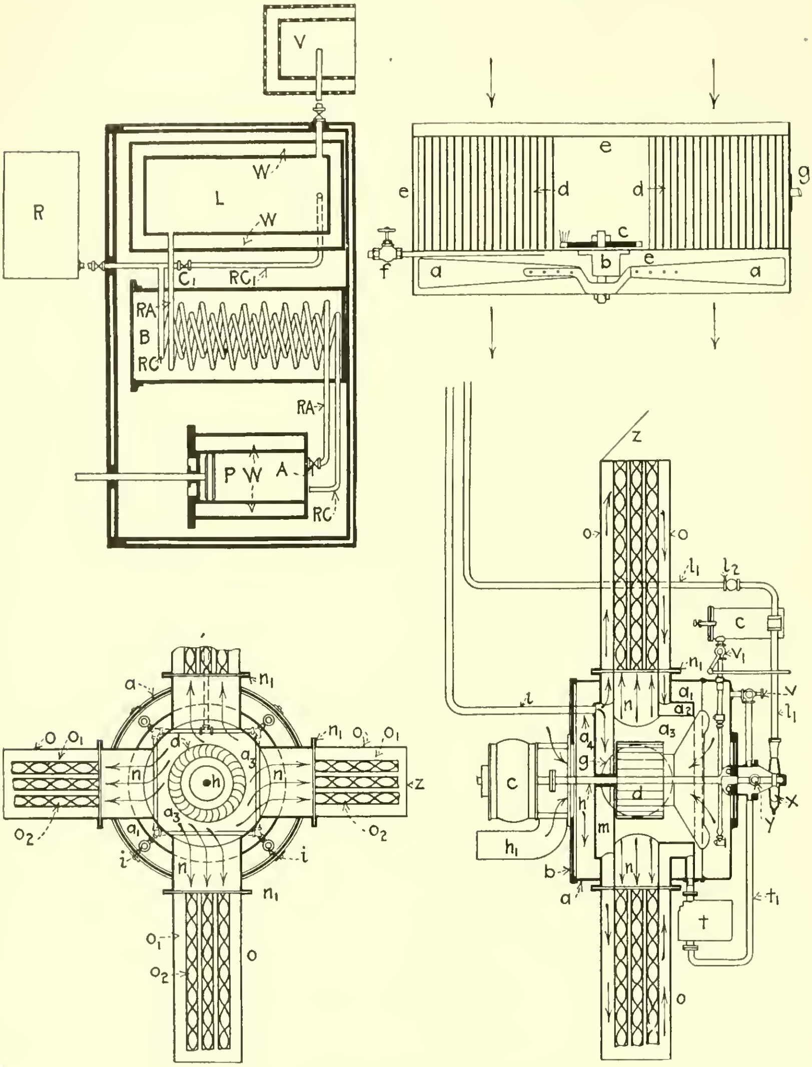

Fig. 1 German Air Conditioning Apparatus

with a heat exchanging medium (ice, steam, etc.), while all the piping and valves are carefully protected against heat losses. If the valves C and C_2 are closed, only the air cooling part is in operation, while the air cleaner is cut out. At the suction stroke of the pump P, with admission valve C_1 open, the air flows through the suction serpentine RC_1 from the reservoir L and is partly cooled in passing through the tank B, while at the same time it takes heat away from the water around L. At the compression stroke of the pump P the exhaust valve A opens, and admission valve C_1 closes, the air is driven through the pressure serpentine RA, gives up heat to the water in the jacket around the pump, and is cooled first by the ice in the tank B, and next by expanding in L. The distributor V consists of two fine meshed wire nets insuring slow and uniform flow of air into the room, this being necessary because strong and non-uniform air currents, particularly of cold air, have an evil influence on the health.

A similar apparatus patented in Germany (patent No. 243341) by Konrad Marquards is *for use in textile plants.* The chamber a has three concentric cavities, the innermost one containing a fan d (Fig. 1B and C). The casing e encloses four water atomizing nozzles i, and supports by its bottom the centrifugal pump x which is on a common vertical shaft with the fan d, and is driven by the electromotor c. The outer circular cavity a_1 serves for the atomization of the water, while the cavity a_2 which forms, by means of a double roof, a second cavity m, serves as a steam space. Through the cavities a_1 and a_2 pass four pipes n starting from the ventilator chamber a_3 and joined at the other end to the four air distributing and heating pipes o. Sections of the pipes n form the connecting ducts q and q_1, the first between the pipes o and steam space m, and the other between these pipes and the steam space a_2. The steam passes through the steam supply pipe r (Fig. 1C) and connecting pipe q, as shown by the dotted lines, into the steam space m, and thence into the air heating pipes o, where it flows around the heating pipes o_1, finally to reach the steam space a_2. The water of condensation moves in the same direction and flows into the water pot t from which it passes through pipe t_1 to the centrifugal pump x and thence either to the atomizing nozzles or to the sewage piping r_1, the water serving for air moistening passing preliminarily through the sieves at s_1. If the air has only to be heated, but no water is to be atomized, the valves v and v_1 are closed, and all the water sent through pipe r. If, on the contrary, it is desired only to moisten the air, but not heat it, the regulating valve r_2 is closed completely. The path of the air in all cases is indicated by the arrows. Outside air is sucked in by the fan d, and passes consecutively through the circular chambers a_4, a_1 and a_3, becoming somewhat warmed up by the contact with the external walls of the chamber a_2. It flows then through the connecting pipes n and heating pipes o_1 into the workshops, so that in winter ventilation and heating occur simultaneously. In ventilation without heating (e.g. in summer) the air is taken in by the fan through the pipe k_1 and let out into the workrooms through pipes o and o_1. The moistening of the air occurs in the chamber a_1

where the air and the water jets from the atomizing nozzles come together.

H. R. Gabler in his patent (No. 243610) combines an *exhaust steam turbine fan drive with heating by the same exhaust steam*, both being done in a single unit. As shown in Fig. 1D, *a* is the fan wheel, *b* the bearing, *c* turbine wheel, *d* air pipes built into the heater, *e* heater, *f* steam supply piping, *g* water of condensation discharge piping. Steam coming through pipe *f* sets into motion the turbine wheel *e* and the fan wheel *a* directly coupled with the latter; while the air driven by suction through the pipes *d* is heated, and at the same time condenses the exhaust steam.

The article contains also a description of the air conditioning apparatus of A. Sauer, of Pittsburgh. It is to be continued.

Mine Fan Drive by Polyphase Motors with Variable Speeds of Rotation (*Antrieb von Grubenventilatoren durch Drehstrommotoren mit regelbarer Umlaufzahl*, Sauvage. *Glückauf*, vol. 48, no. 41, p. 1668. October 12, 1912. 3 pp., 4 figs. *ep*). At the Hermine mine in Neunkirchen (Trier, Germany) was installed a fan plant designed to supply at the time of its installation 5600 cbm (say 200,000 cu. ft.) of air per min., but to be able to supply later 8000 cbm (say 282,000 cu. ft.) per min. A Rateau fan of the latter maximum capacity was installed, with a speed varying in accordance with the output desired between 150 and 220 r.p.m. Three-phase current was available at 2000 volts and a frequency of 50. The article describes in detail the somewhat unusual electric equipment of the plant and its testing. During the tests three methods of regulation were applied: by a regulating starter, by varying resistances with asynchronous motors, and by throttling the suction piping of the fan. Fig 2A gives a comparison of the first two methods, showing, by the way, that at low speeds and correspondingly low fan outputs change from Delta connection to Star connection improves the efficiency. Fig. 2B shows the power consumption at various outputs with all three methods of regulation, proving, as might be expected, that for all but the very highest outputs, regulation by throttling is the least economical.

In the present case the first cost of the regulating apparatus increased the cost of the motors by M.15000 (roughly, $3700). If the installation were driven during the first 16 months at an average speed of 340 r.p.m., or at 70 per cent of maximum air output. there would be a saving of 53 kw. during that time, or, at 8500 hours per year and at 2.5 pf. (say $0.006) per kw-hr., there would be in those first 16 months a saving of 16×8500×53×2.5 ÷ 12×100=M 15.016.66, or the extra first cost would be saved in less than a year and a half.

A New Blower (*Ein neues Geblase*, J. Lehne. *Die Turbine*, vol. 9, no. 1, p. 11. October 5, 1912. 2 pp., 2 figs. *d*). After discussing the usual disadvantages of blowers using an auxiliary liquid the author describes a new blower of the International Rotary Machine Company of Berlin using a constantly rotating ring of liquid *e* enclosed, to avoid excessive friction losses, in a freely revolving drum *b* (Fig. 3B), whereby the small friction at the bearings is substituted for the considerable frictional resistances at high velocities. Eccentrically to the axis of this drum is placed a blade disc *a* enclosed by side plates as shown, touching the ring of liquid at one place of its inner circumference while everywhere else there is a certain distance between the edges of the blades and the circumference of the ring varying from zero to maximum and back to zero. As the drum is very rotatably placed in the ball bearings *d* and *e* (Fig. 3A), it is easily carried away in its rotation by the ring of liquid, which thus forms an air-tight enclosure of the separate cell spaces. Inside the wheel *a* is a fixed wrist plate *f* with a suction duct *g* and pressure duct *h*, the latter so proportioned that it is set in connection with the corresponding cell-openings only after the pressure in the respective cells has reached exactly that level for which the apparatus is designed.

Tests with small machines having a wheel with inner diameter of 80 mm (3.14 in.) have shown an efficiency of 60 to 73 per cent when the machine worked as a blower, and 50 to 63 per cent when as a vacuum pump. Pressures up to 1.2 atmospheres above atmospheric have been obtained with water used as the auxiliary liquid, but with mercury as the auxiliary liquid proportionally (to its specific weight) higher pressures may be obtained. The above described arrangement may be used therefore for obtaining considerable pressures with small sized machines.

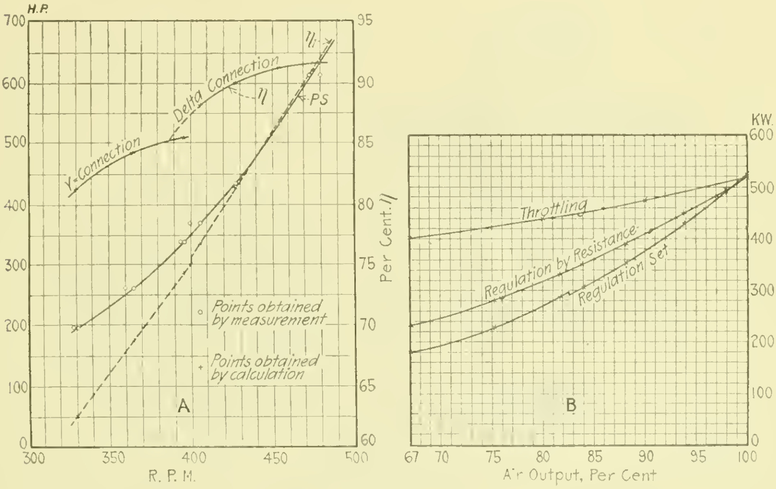

FIG. 2 MINE FAN DRIVE BY POLYPHASE MOTORS, EFFICIENCY AND ECONOMY OF REGULATION

Internal Combustion Engines

INTERNAL-COMBUSTION ENGINES IN THE OIL INDUSTRY (*Die Verbrennungskraftmaschinen in der Erdöl-Bohrindustrie*, Neumann. *Zeits. des Internationalen Vereines der Bohringenieure und Bohrtechniker*, vol. 19, no. 20, p. 233. October 15, 1912. 5 pp., 11 figs. *d*). From a paper by the author read before the International Society of Drilling Engineers and Machinists in Berlin, on the application of internal-combustion engines in drilling for oil. The author points out the advantages of having engines which can work equally well on natural gas and liquid fuel, and describes in detail the Deutz engine which satisfies this condition.

Power Production in Gasworks (*Die Krafterzeugung in Gaswerken*, A. Krauss. *Journal für Gasbeleuchtung*, vol. 55, nos. 37 and 38, pp. 901 and 925. September 14 and 21, 1912. 11 pp., 7 figs. *gp*). The author claims that German gas works do not make sufficient use of coke breeze and other fuel refuse which, as he shows, may be well used for the production of power; he quotes the American practice, and discusses in detail the McLean process.

A Six-Cylinder V-Motor at 60 Deg. (*Le moteur a 6 cylindres en V a 60°. La Technique automobile et aérienne*, vol. 7, no. 82, p. 153, October 15, 1912. 3 pp., 4 figs. *t*). Theoretical investigation of a six-cylinder motor in V with the arms of the V at 60 deg., its order of ignition, and balancing.

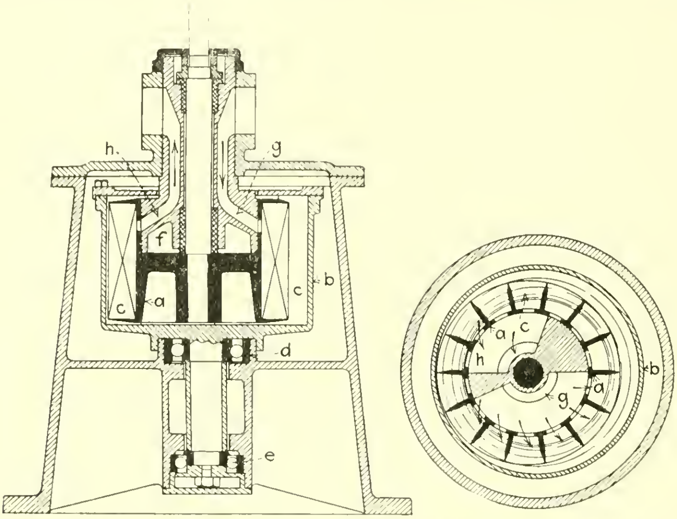

Fig. 3 Blower of the International Rotary Machine Company of Berlin

Modern Progress and Experiences in the Technical Application of Coal Tar Derivatives for the Production of Heat, Power and Light. (*Neuere Fortschritte und Erfahrungen in der technischen Verwendung der Teerprodukte für Heiz-, Kraft- und Lichtzwecke*, A. Dahm, *Zeits. für angewandte Chemie*, vol. 25, no. 40, p. 2049. October 4, 1912. 9 pp., 14 figs. *dg*). A general discussion of the application of coal tar products and the use of coal tar itself as a fuel, with the description of some of the standard apparatus used in this connection.

The Gas Turbine (*Über die Gasturbine*, Hans Holzwarth. *Journal für Gasbeleuchtung*, vol. 55, no. 39, p. 949. September 28, 1912. 7 pp., 14 figs. *dl*). Description of the Holzwarth gas turbine (cp. The Journal, February 1912, p. 303), and some controversy with Professor Stodola (cp. The Journal, September 1912, p. 1384). The author points out that if steam is used in gas turbines to reduce the temperature of the explosive mixture, as proposed by Davey (*The Engineer*, March 8, 1912 and ff.), the sulphur derivatives in the gaseous mixtures will pass into hydrates, and form compounds powerfully attacking the metal parts of the turbine, and it is practically impossible to regulate the injection of water with such a precision as to have in the turbine at all times only superheated steam. As to the use of convergent-divergent nozzles in gas turbines, the author states that his experiments have shown that in most cases such a nozzle is entirely unadapted for use in gas turbines; gas turbines do not have nozzles in the strict sense of the term, but use single plane parallel outlet openings of which the efficiency is independent of the pressure fall, and superior to that of a nozzle. As to the supposition that the gas turbine wheel has little efficiency owing to the variable velocity of the gas jet, the author states that his tests have shown that it is not materially below that of the steam turbine, for although the decreasing velocity of the jet is theoretically an unfavorable factor, there must be some other factors which counteract its influence.

As to the possibility of the gas turbine in its present state of development being commercially applied, the author agrees that the peak values of its efficiency curve are below those of some other prime movers, but points out that the commercial applicability of an engine is not determined by the peak values of its curve, but by the overall *commercial* economy of application. At full load the reciprocating gas engine is nearly twice as efficient as the steam turbine, but the latter is widely applied simply because the usual loads are only about 50 per cent of the maximum (this applies to steel plants only). The gas turbine, as has been shown by the tests, may use gas direct from the scrubber, without passing it through a dry cleaner, and without injuring thereby the gas admission valves; there is some deposit of dust and small particles of coal in the combustion chamber, but since no oil reaches the combustion chamber, this deposit does not cling to the walls, as in the case of reciprocating gas engines, and generally gives no trouble. No corrosion or erosion has been observed at the nozzle valves; the nozzles and blades become rapidly covered with a hard and resisting skin of iron oxide-oxidul, and after that are no more attacked by the hot gases. As to the overall commercial economy, the author calls attention to the fact that amortization, interest, attendance, etc., in steel plants, has been found to be 55 per cent of the cost of fuel in the case of steam turbines, and 245 per cent in that of reciprocating gas engines, and the author believes that for gas turbines the steam turbine ratio will hold. He states further that his gas turbine can be driven, in addition to gas, with the same liquid fuels as the Diesel engine.

Gas Turbines (*Turbing plynove*, J. Dejmek. *Technicky Obzor*, vol. 20, no. 30, 31, and 32, pp. 221, 229 and 237, September 18, October 2 and 9, 1912. 9 pp., 17 figs. *dtA*). The author makes the usual distinction between combustion and explosion turbines, and investigates the efficiency and characteristics of both Every gas turbine requires a compressor which takes up the theoretical amount of work L_c, or the actual amount of work $\frac{L_c}{\eta_c}$ where η_c is the efficiency of the compressor. If L_t be the theoretical output of the turbine without compressor, and L_z that with the compressor, then

$$L_z = L_t - L_c$$

If, further, η_t be the efficiency of the turbine without compressor, the effective work of the turbine without compressor is $\eta_t L_t$, and that of turbine with compressor

$$L_c = \eta_t L_t - \frac{L_c}{\eta_c}$$

while the overall efficiency of a turbine with compressor is

$$\eta = \frac{L_c}{L_z} = \frac{\eta_t L_t - \frac{L_c}{\eta_c}}{L_t - L_c} = \frac{\eta_t - \frac{L_c}{L_t} \cdot \frac{1}{\eta_c}}{1 - \frac{L_c}{L_t}}$$

In connection with the combustion gas turbine the author investigates the following two processes: (*a*) adiabatic compression and adiabatic expansion, and (*b*) isothermic compression and adiabatic expansion. In the following let T_1 be the absolute temperature at the end of compression, T_2 at the beginning of expansion, T_3 at the end of expansion, and T_o at the beginning of compression. Since the amount of heat and work brought in must be equal to that of heat and work taken away

$$L_t - L_o = Q_1 - Q_2$$

where Q_1 is the amount of heat brought in, and Q_2 that escaping through the exhaust, whence

$$Q_1 = c_p\,[T_2 - T_1], \quad \text{and} \quad Q_2 = c_p\,[T_3 - T_o]$$

The absolute heat efficiency is

$$\eta_a = \frac{Q_1 - Q_2}{Q_1} = \frac{c_p[T_2 - T_1] - c_p[T_3 - T_o]}{c_p[T_2 - T_1]} = \frac{T_2\left(1 - \frac{T_1}{T_2}\right) - T_3\left(1 - \frac{T_o}{T_3}\right)}{T_2\left(1 - \frac{T_1}{T_2}\right)}$$

but since in adiabatic compression or expansion

$$\frac{T_1}{T_2} = \frac{T_o}{T_3}$$

hence

$$\eta_a = \frac{T_2 - T_3}{T_2} = 1 - \frac{T_3}{T_2} = 1 - \frac{T_o}{T_1}$$

which shows that the heat efficiency depends on the temperature ratios and not on the maximum temperatures used in the process. It is further shown that the following must hold:

$$\frac{L_c}{L_t} < \eta_t \cdot \eta_c$$

since otherwise no work would be produced. But the following equation is also correct:

$$\frac{L_o}{L_t} = \frac{T_1 - T_o}{T_2 - T_3}$$

and since for the case of adiabatic compression or expansion

$$\frac{T_1}{T_2}=\frac{T_0}{T_3}$$

therefore

$$\frac{L_c}{L_t}=\frac{\dfrac{T_0}{T_3}-\dfrac{T_0}{T_2}}{1-\dfrac{T_0}{T_1}}$$

which, for the usual range of temperature, may be written with sufficient correctness in the following form:

$$\frac{L_c}{L_t}=\frac{T_0}{T_3}$$

This shows that the chief condition for the production of any work at all is that

$$\frac{T_0}{T_3}<\eta_t\,.\,\eta_c$$

Taking a numerical example, and substituting for η_c0.8, for η_t 0.7, and for T_0 300 deg. cent., the following must hold

$$\frac{300}{T_3}<0.7\times 0.8$$

or

$$T_3>535 \text{ deg. cent.}$$

Process of Isothermic Compression and Adiabatic Expansion. In this process $T_0=T_1$. The loss of heat in the process with isothermic compression is expressed by the equation

$$Q=Q_2+Q'_2$$

where Q_2 is the heat lost through the exhaust, and Q'_2 that lost through compression, the latter being expressed by the equation:

$$Q'_2=ART_0\,\mathrm{l_{nat}}\,\frac{P_1}{P_0}$$

Substituting this value in the equation for the absolute thermal efficiency

$$\eta_a=\frac{c_p(T_2-T_1)-c_p(T_3-T_1)-ART_1\,\mathrm{l_{nat}}\,\dfrac{P_1}{P_0}}{c_p(T_2-T_1)}$$

But as to the effective work of the turbine, the following must hold:

$$\frac{L_c}{L_t}<\eta_c\,\eta_t$$

Since, however, it was shown above that

$$L_c=RT_0\,\mathrm{l_{nat}}\,\frac{P_1}{P_0}$$

and

$$L_t=R\,.\,\frac{k}{k-1}[T_2-T_3]$$

therefore

$$\frac{L_c}{L_t}=\frac{k-1}{k}\,\frac{T_0}{T_3}\,\frac{\mathrm{l_{nat}}\,\dfrac{P_1}{P_0}}{\dfrac{T_2}{T_3}-1}$$

Substituting corresponding values from equation

$$\left(\frac{T_2}{T_3}\right)^{K} = \left(\frac{P_1}{P_o}\right)^{K-1}$$

the following is finally obtained

$$\frac{k-1}{k} \cdot \frac{T_o}{T_3} \cdot \frac{l_{nat}\frac{P_1}{P_o}}{\left(\frac{P_1}{P_o}\right)^{\frac{k-1}{k}} - 1} < \eta_c \cdot \eta_t$$

which shows that with constant T_3, T_o is a function of the pressures p_1 and p_o.

The author treats in the same manner the explosive gas turbine, this part being omitted partly owing to lack of space, partly because the theory of the explosive turbine is considerably better known through the publications of Holzwarth, Davey, H. H. Suplee, and others.

The article further discusses the heat entropy diagrams for the gas turbines, and describes the Holzwarth turbine.

Diesel Engine Building at the Germania Shipbuilding Yard (*Der Dieselmotorenbau auf der Germaniawerft*, C. Regenbogen. *Die Gasmotorentechnik*, vol. 12, no. 7, p. 113. October 1912. 3½ pp. *gh*). A historical sketch of Diesel engine building at the Germania shipbuilding yard, one of the largest in Germany, and some considerations on the design, mainly that of large Diesel engines for marine purposes. The problem of *cooling the piston head* was at first solved by using as a cooling medium lubricating oil, because if it should happen to leak into the cylinder, it could do no particular harm there, as would water, particularly sea water. It was found, however, after a series of comprehensive tests that in the case of very large two-stroke cycle motors water as a cooling medium gives better results when carefully selected and suitably designed apparatus is used for the distribution of the liquid.

Experiments have shown that double-acting Diesel engines may now be built without much difficulty at 2000 h.p. up per cylinder. In fact, the firm has constructed at considerable expense a large two-stroke cycle double-acting stationary single-cylinder engine with scavenging pump at the side of the main cylinder, and it proved a success. As regards the improvement of specific efficiency of two-stroke cycle engines, the limits of piston velocity have been practically reached, and further progress may be expected only in improvements of the compression of gas mixture in the cylinder and still better atomization of fuel.

The Micro-Indicator (*Der Mikro-Indikator*, O. Mader. *Der Motorwagen*, vol. 15, no. 17, p. 632. September 30, 1912. 4 pp., 19 figs. *d*). Description of a special *indicator* invented by the author *for use with high-speed small internal-combustion engines*, where the ordinary indicator cannot well be used. Attempts have been made to use optical indicators for pressure recording but their disadvantage is that their record must be made photographically, and that leads to the diagram lines being of very uneven distinctness, and being underexposed in fast moving machinery, and overexposed in indicating slow-speed engines. The author's *micro-indicator*, like the early steam indicators, has a short piston stroke and direct recording, in this case, however, made with a sharp steel point on a soot blackened glass; the fine lines are then magnified under a microscope, outside of the engine installation proper. Fig. 4A schematically represents the gear which allows the indicator record to be made by minute sidewise motions of the light steel point. The piston rod has at the top a flat plate *a* which is in gliding contact with the end of the rod *b*. The point *s* of the recording style lies exactly in the axis through the center of the circular end of the rod *b* of which the second end *E* is actuated by the lever *c* swinging about *D*. To keep the rod *b* always in gliding contact with the end of the piston rod, and to avoid lost motion in the gear, the many-coiled spring *f* is introduced. Since many guide points, even with the best adjustment, are apt to produce excessive friction, the parts in this indicator are all guided at no more than two points; thus, the piston is guided below by the cylinder wall, above by the double wound spring, the spring, rod and piston being combined in one piece, and, should another scale be required for the diagrams, must all be changed together; this, however, eliminates all

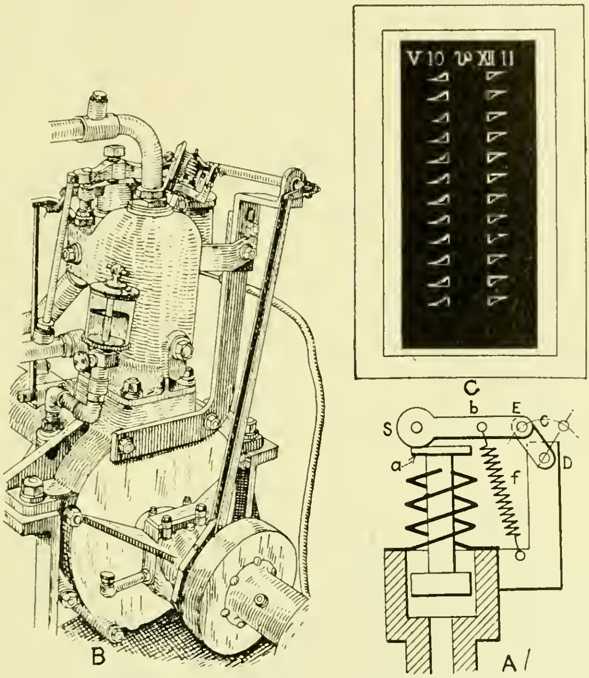

Fig. 4 Mader Micro-Indicator, Its Drive and Diagrams in Natural Size

possibilities of their getting loose in operation or being incorrectly put together at the beginning.

An important part of the micro-indicator is the diagram holder, a U-shaped frame rotatable about a fixed axis; the soot blackened glass is slipped in from above and held fast by a spring. The simplest drive, with as few joints as possible, is used, e. g. a rocking drive (Fig. 4B). To avoid large errors in the diagrams, the setting of the drive must be carefully made not only as regards dead centers, but also with respect to the average piston position at which the recording style is instantaneously at rest previous to changing its direction of motion.

The diagrams are magnified by an ordinary microscope with a 1:40 rate of magnification, which is sufficient for an estimate of the diagram by inspection. If measurements are to be made, the diagrams are magnified either photographically, or drawn to a larger scale by means of a special reflecting and magnifying device.

The micro-indicator may be used for the determination of the indicated horsepower of a high-speed internal-combustion engine (600 to 3000 r.p.m.); the glass on which the records are made may be cleaned, covered by a new layer of soot or lampblack, and used over and over again.

INFLUENCE OF THE SPEED OF COMBUSTION ON THE EFFICIENCY OF A GAS ENGINE (*Influence de la vitesse de combustion sur le rendement d'un moteur à gaz*, E. Mérigeault, *Comptes Rendus des séances de l'Académie des Sciences*, vol. 155, no. 15, p. 644. October 7, 1912. 3 pp., 3 figs. *t*). The author establishes a theory of the gas engine which may be applied independently of the nature of the gaseous medium and of its molecular contraction produced by combustion. He starts from a previously established equation

$$AT_u = L_v - \quad - \int_{T_B}^{T_E} c\,dT - p_a(V_E - V_B)$$

which is true whatever the nature of combustion, and in which T_u is the work produced by the engine during a complete cycle (two revolutions); L_v the calorific value, at constant volume, of the combustible admitted during one piston stroke; q the quantity of heat given up to the walls during the time between the end of the admission B to the beginning of the exhaust E; T' and T_E the temperatures, supposedly uniform, at those two instants; c the specific value, at constant volume, of the whole mass of gases contained in the cylinder at the end of combustion; E_B and V_E volumes at the instants B and E respectively. The author proceeds to prove the following theorem (modification of Carvallo's, but applicable to all gases): If there be two motors having walls impermeable to heat (i.e. with $q = 0$), and with volumes V_E and V_B equal respectively, absorbing at each stroke the same quantity of combustible gas, and *working between the same limits of pressure;* if in one of these two cylinders the combustion is at constant pressure, while in the other it follows some other law, but is complete at the moment of exhaust, *the first has a better efficiency.* Let *BCDE* and *BC′ D′ E′* (Fig. 5) be the diagrams of these two motors, where *DE and D′ E′* are two adiabatic curves of the same gaseous system (completely burned gas). The author first proves that E' cannot be above E, as in the diagram on the left, and that it must therefore be below it, as on the other diagram. Hence $T_E^1 > T_E$ and consequently

$$\int_{T_B}^{T_E'} cdT > \int_{T_B}^{T_E} cdT$$

But from the assumption of equal limits of pressure in the theorem, $C' D'$ must be tangent to CD, and

$$AT_u = L_v - \int_{T_B}^{T_E} cdt - Ap_a\,(V_E - V_B)$$

$$AT_u - L_v \int_{T_B}^{T_E'} cdT - Ap_a\,(V_E - V_B)$$

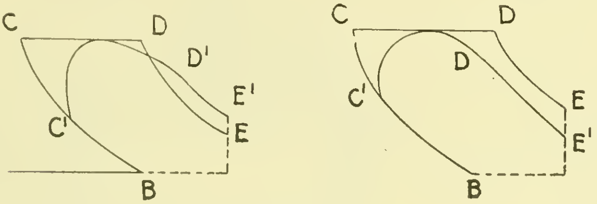

Fig. 5 Indicator Diagrams

and from this and the preceding inequality is deduced

$$T_u > T'_u$$

which shows not only the comparative values of the two efficiencies, but also the relative position of the two diagrams.

Mixed Combustion-Motor Sabathé (*Motore à combustione mista Sabathé. L'Industria*, vol. 26, no. 41, p. 645. October 13, 1912. 3 pp., 8 figs. *d*). Description of the Sabathé engine, double-acting, four-stroke cycle, of the *mixed combustion* type, i.e. with the admission of the fuel, after the compression of the air, so regulated that combustion occurs first at constant volume, and then at constant pressure. In an explosion motor (Fig. 6 L) the heat which has to be transformed into work is produced instantaneously in the cylinder, and there is an instantaneous rise of pressure AB at constant volume. In a slow combustion motor (Fig. 6 M) the production of heat takes place during a quite appreciable length of time, and at about the same rate as it is transformed into work. the temperature of the fluid remaining therefore nearly constant; the period of combustion is shown by AB, the working process being that of the Diesel engine. The *Sabathé engine* follows a mixed process (Fig. 6 N) : part of the heat is produced in AB at constant volume, with rise in pressure and temperature, and part in BC at constant pressure. Theoretically the Sabathé motor is superior or at least equal to the Diesel engine, as the author shows by an analysis of its thermal diagram.

The most interesting part of the engine, the needle device which helps to produce mixed combustion, is shown in Fig. 6, O: the needle *A* has a conical tip fitting into a conical orifice in a bush held at the bottom of the cylinder by springs; a cam at a convenient moment lifts the needle *A* which in its turn raises the valve *S*, by means of the wipers *T*, when it has gone a certain distance in its upward motion. The fuel passes through the passage *B* and at first fills the chamber *G* below the valve *S;* the compressed air for the atomization of the fuel arrives by the duct *D* and reaches the chamber *G* through a groove cut along the valve *S*. The working process is as follows: at low speed, the fuel which nearly fills the chamber *G* is injected into the cylinder when the needle moves upward and a little before it reaches the neutral position, and explodes at constant volume; when the output of the engine reaches a certain amount, the chamber *G* cannot hold all the fuel that is required, and some of it accumulates in the chamber *E*. Therefore, when the needle is lifted, first the fuel in *G* is projected into the cylinder, and explodes at constant volume, and then, after the wipers *T* have raised the valve *S*, the fuel from *E* is admitted into the cylinder, and is consumed there in accordance with a law depending on the shape of the cam, and nearly achieving combustion at constant pressure. For the atomizer proper of the Sabathé engine see H. R. Setz, Oil Engines, Trans. AM.Soc.M.E., vol 33, 1911, p. 868.

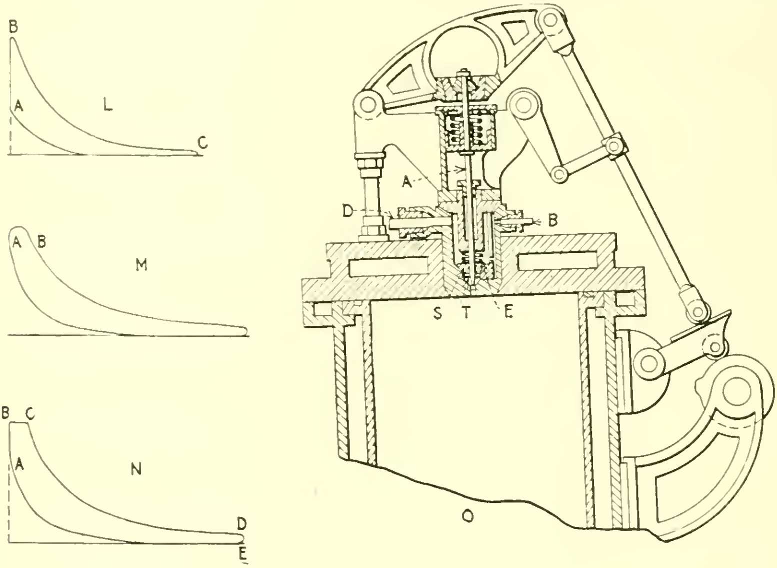

Fig. 6L, M, N Diagrams of Various Types of Internal-Combustion Engines
O Tappet Gear of the Sabathe Engine

The Sabathé engine can use any kind of liquid fuel, from petroleum to the heavy and cheap oils, and consumes, for large motors, less than 0.2 ng.

(0.44 lb.) per effective h.p., and in small motors from 230 to 0.24 ng. (0.50 to 0.53 lb.).

A NEW FREIGHTER EQUIPPED WITH DIESEL ENGINES (*Un nuovo piroscafo da carico con motori Diesel*, G. Supino. *L'Industria*, vol. 26, no. 40, p. 631. October 6, 1912. 3½ pp., 13 figs. *d*). Description of the Diesel engine freight carrier "Monte Pinedo," 4000 register tons. For a complete description see *Motor Boat*, October 10, 1912, p. 10.

Machine Shop

THE OXY-ACETYLENE WELDING BURNER, ITS OPERATION AND CONSTRUCTIONAL REQUIREMENTS (*Der Acetylen-Sauerstoffschweissbrenner, seine Wirkungsweise und seine Konstruktionsbedingungen. Acetylen*, vol. 15, no. 16, p. 145, in the pagination of the appendix *Autogene Metallbearbeitung*, 5 pp., 3 figs. *ct*). Abstract of a doctor's thesis under the above title presented by Hans Ludwig at the Technical High School of Berlin, and covering besides general considerations on the theory of burners for autogenous welding, the description of the test arrangements, calibration of instruments, and test data. The following are the most important conclusions of the author's investigation: (*a*) the gas mixture in the burner should consist at the moment of combustion of equal percentages of oxygen and acetylene, while the velocity of the flow of gas must vary, in accordance with the type of burner, from 80 to 140 m (262 to 460 ft.) per sec.; (*b*) the cross-sections of the passages in the burner must be adjustable to make possible an admission of an excess of acetylene in order to regulate the ratio of gas mixture which may be changed through the heating of the tip of the burner; (*c*) the preheating of oxygen produces increase in the velocity of the flow of the gas, and decrease of the mass of oxygen flowing through the nozzle per unit time; (*d*) the tests have established the fact that the mixture of the gases at the burner nozzle produced by a simple mixing chamber is perfect, and there are no separate streams of oxygen and acetylene; (*e*) the temperature of the welding flame is estimated as between 3000 and 3400 deg. cent. (5432 to 6152 deg. fahr.). The experiments have further shown that with low-pressure burners the mixture ratio may be varied by regulating the oxygen supply only, since, although the acetylene supply may be reduced by throttling, it cannot be always increased as desired. As to the velocity of flow of gas through the nozzle it was found that it is right when there is no back firing during the welding; the particular velocity for each burner has to be established by trials, within the limits set above.

The tests have established a complete absence of standards for regulating the gas consumption of burners per unit of work done, and some burners consume nearly 50 per cent more gas than others for the same amount of work done; with the existing difficulties of measuring the temperature of the flame it is, however, difficult to determine the actual efficiency of the different types. The author discusses in detail the influence of various proportions of passages in the burner on its efficiency, and recommends certain ratios.

TESTS OF AUTOGENOUS WELDING (*Essais sur la soudure autogène*, Desjuzeur, La Houille Blanche, vol. 11, no. 8, p. 201. August 1912. 4 pp. *e*). The article is based partly on the old tests of Professor Bach of Stuttgard, partly on recent tests made for the author, who is director of the Lyons Association of Owners of Steam Machinery, by Professor Bach and the laboratory of the Steel and Foundry Company of Firminy, on test pieces supplied by the Dissolved Acetylene Company of Marseille, France. These tests have shown that a well-made weld does not create a weak point as far as resistance to static effort is concerned. In some cases where the excessive thickness produced at the welded spot has not been planed off, the rupture occurred elsewhere; where it has been removed by planing, the rupture occurred at the weld, but the stress in all cases has been but slightly below the usual breaking stress for the given metal. The elongations, however, have been markedly reduced, 39 to 47 per cent for bars with excessive thickness at the weld not planed off, and 67 per cent for planed-off bars. Tension tests at 200 deg. cent. (392 deg. fahr.) have shown that at this temperature the elongation in welded sheets is still below that of the unwelded metal, and it would be dangerous therefore to accept, as some do, that the ductility of the metal in case of boilers is improved by the heating to which the sheets are subjected. It was further found that, probably owing to the action of rapid cooling at the welded joint, the metal there is considerably harder than elsewhere, and that the welded joint has a lower resistance to shocks. The resilience at the weld is approximately only one-seventh of that of the unwelded part of the sheet, which the author ascribes, from data furnished by the chemical analysis, to the presence of a large amount of phosphorus in the welded joint, which may again be due either to the quality of the iron in the melted bar, or to the presence of hydrogen phosphate in the acetylene.

From a microscopic investigation and other data the author expressed a conviction that it would hardly be possible at the present time to have obtained better autogenously welded joints than those which were submitted to his tests. It is therefore difficult to say what may be the strength of the less carefully made joints, especially in jobs on marine boilers where the welder works under extremely unfavorable conditions, lying on his back, cramped as to space, and all the time exposed to the intense heat from the burner.

USE OF CONCRETE IN MACHINERY FOUNDATIONS (*Anwendung von Beton zu Maschinenfundamenten, Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 38, p. 1546. September 21, 1912. 3 pp., 8 figs. *dp*). Supplied by the Information Bureau for Concrete Constructions in Heidelberg, Germany. The article describes how concrete foundations may be changed with the change of machinery, partly using the old foundations for the new purpose. A good plan is to drill holes by a compressed air drill, and then break the concrete by hand bars, or to remove the concrete by explosives. The broken material may be easily removed by hand and used again. The article also gives detailed instructions as to the concrete mixture and manner of laying the foundation.

MAKING PATTERNS FOR SINGLE AND DOUBLE AUTOMOBILE CYLINDERS (*Anfertigung von Modellen zu einfachen und doppelten Automobilzylindern*,

J. Biller. *Werkstattstechnik*, vol. 6, no. 20, p. 525. October 15, 1912. 4 pp., 13 figs. *p*). Detailed instructions for making patterns for automobile engine cylinders, with data on the cost of same in Germany.

CRYSTALLIZATION BY REHEATING OF THIN DRAWN METAL (*Cristallisation par recuit des métaux écrouis*, F. Robin. *Comptes Rendus de l'Académie des Sciences*, vol. 155, no. 13, p. 585. September 23, 1912. 2 pp. *e*). The author has found by experiments reported in the article that when a thin drawn metal is reheated, the size of its crystals is a function of the temperature and duration of reheating, but with the same duration of reheating, the *size of the crystals does not grow regularly with the temperature of reheating*, since neither the largest grains are obtained with the highest temperatures of reheating, nor the smallest grains with the lowest temperatures. Generally the grains which are large in the proximity of the temperature of melting, at first decrease rapidly with the temperature, and then remain stationary for a time. Impurities have an important influence in reducing (except in the case of zinc alloyed with copper) the size of the crystals without materially affecting the temperature at the beginning of reheating.

PROCESS FOR THE MANUFACTURE OF IRON AND STEEL AND FURNACES FOR USING IT (*Procédé pour la fabrication du fer et de l'acier et four pour la mise en pratique. La Métallurgie*, vol. 44, no. 40, p. 613. October 2, 1912. 1 p., 7 figs. *d*). A new process for the manufacture of iron and steel patented by Meyer Davidsen. The disadvantage of the regenerative furnace consists in the heat losses involved in the regeneration process, and the necessity of varying the direction of the flow of air frequently. *In Davidsen's process finely powdered coal is burned above the metal bath.* All the heat contained in the coal is instantly developed in the very laboratory of the furnace, and all losses in producing the gas in gas producers and preheating it are entirely eliminated. A further advantage of the new process consists in the method of communicating the heat to the metal. The flame produced by burning finely ground coal consists of an extremely large number of minute incandescent particles having a high radiating capacity; there is therefore in the first instance no need to bring the surrounding air to the temperature of the coal particles, the heat being communicated by radiation, and not by convection, and in the second place the flame may be kept high enough above the metal bath to prevent its contamination by impurities from the coal, and that permits of doing away with the arch which at best is only a source of trouble and expense. The temperatures obtained in such furnaces are claimed to be very high, closely approaching those of the electric furnace. By the elimination of frequent reversals of the direction of the flame various parts of the furnace are kept always at the same temperature. The rate of the flow of air must be large enough to prevent deposits of ash in the furnace proper. The article gives the details of the furnace construction; it is to be continued.

SAND-BLASTING MACHINES AND THEIR APPLICATION TO CLEANING IRON AND OTHER STRUCTURES (*Sandstrahlgebläse und deren Anwendung zur Reinigung von Eisenkonstruktionen und sonstigen Bauwerken*, W. Eckler. *Der*

Eisenbau, vol. 3, no. 10, p. 372. October 1912. 6 pp., 7 figs. *deh*). A general and historical sketch of the *sand-blasting machine practice*. The author quotes data of tests which show that sand-blasting is superior to steel brushes, or to repeated bending for cleaning of iron. As to the efficiency of compressed air sand-blasting machines, he gives the following information. With average pressure of 1 to 1.2 atmospheres, and good sharp sand the following areas per hour have been cleaned:

Diameter of Nozzle.		Area Cleaned.	
Mm	In.	Qm.	Sq. ft.
6	0.236	2 to 3	21.5 to 32.3
10	0.393	4 to 6	43.0 to 64.5
16	0.629	10 to 15	107 to 160

Cp. also Wm. T. Magruder, Tests of a Sand-Blasting Machine, Trans. Am. Soc. M. E., 1911, p. 821.

Mechanics

Theoretical Conditions for the Installation of Change of Speed (*Conditions théoriques de l'établissement d'un changement de vitesse*, G. Lienhard. *La Technique automobile et aérienne*, vol. 7, no. 81, p. 132. September 15, 1912. 3 pp., 4 figs. *mt*). A theoretical investigation of the principles of *speed changing*, mainly with regard to automobile engines. The author comes to the following conclusion: the car must be so constructed and *nn'* (reduction due to the change speed gear box times reduction due to the pinion and crown wheel of the differential) so selected that the speed of the motor should correspond to its maximum capacity while the torque should be but slightly below its maximum value. With this condition satisfied, within its normal range the motor will use its maximum capacity at the highest efficiency.

A Study of Cardan Shaft Drives (*Etude sur les transmissions par cardan*, A. Contet. *La Technique automobile et aérienne*, vol. 7, no. 81, p. 135. September 15, 1912. 3 pp., 5 figs. *t*). The author investigates various types of *Cardan drives*, and finds that the single joint Cardan drive, though apparently unsatisfactory, is really not so bad. When the cardan shaft is long, the joint works at small angles, and with an elastic shock absorber suitably placed, may give very good results; the above mentioned precautions reduce the wear of the tires due to speed variations to a minimum, while the simplicity and ease of operation of the simple cardan drive cannot be neglected. The author further examines attempts to compensate the speed variations due to the first cardan joint by introducing a second one with equal and opposite variations. Such a compensation is absolutely perfect when the driving and driven shafts are strictly parallel, while the intermediate shaft has both its pinions in the same plane (Fig. 7A): the original and the compensating curves are similar throughout though of opposite signs, and the resultant is therefore the diameter of the circle itself. Such an arrangement has been tried on an automobile by Mildé-Gaillardet in France, but it was found to be too complicated in execution; besides, the large number of joints favored the appearance of play, and that produced chattering. Still less satisfactory was the solution when it was attempted to produce compensation of the velocity variations simply by having the two extreme shafts parallel with each other, but making with the intermediate shaft an angle β (Fig. 7B), with the pinions of the intermediate shaft in two planes normal to each other. In this figure , is the angular velocity of the driving shaft, ω_2 that of the driven shaft, ω' that of the intermediate shaft, the ratio $\frac{\omega}{\omega'}$ is expressed by the continuous thin line, the compensating ratio $\frac{\omega'}{\omega_2}$ is represented by the dotted line, while the heavy line shows the curve obtained by summing the ordinates of both curves. It has a large amplitude, and three angular points which indicate a tendency towards shocks and rapid wear of the tires on the car. The author investigates in a similar manner the cases when each of the shafts makes an angle β with the intermediate shaft, the total angle between the driving and driven shafts being 2β: if the pinions of the intermediate shaft are in the same plane, the curve shown in Fig. 7C, is obtained: in its major part it is perfectly flat and coincides with the diameter of the circle, but it has the disadvantage of having three points of angular variation where sudden changes of velocity occur; when the planes of the pinions are normal to each other, the ordinates are simply superimposed, or the ordinate of the final curve is twice that of the original, and not only is there no compensation, but all the bad effects of the original arrangements are doubled. The author believes, however, that with long shafts and small angles, the arrangement shown in *C* may work fairly well.

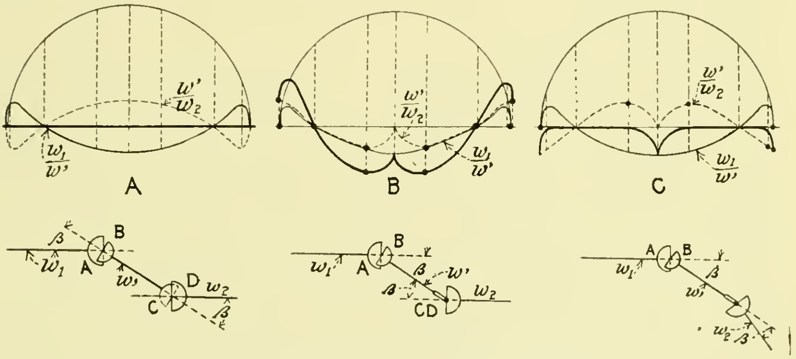

Fig. 7 Cardan Drives of Various Types

New Bicycle Bearing (*Ein neues Fahrradlager*, W. M. *Der praktische Maschinen-Konstrukteur*, vol. 45, no. 20, p. 344. September 26, 1912. 2 pp., 2 figs. *d*). Description of the new bicycle ball bearing, without cones, used by the Deutsche Waffen- und Munitionsfabriken in Berlin, Germany.

Reversible Propellers for Large Outputs (*Umsteuerschrauben für grosse Leistungen*, W. Helling. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 37, p. 1485. September 14, 1912. 4½ pp., 15 figs. *dt*). Investigation of the theory of reversible propellers, and difficulty of their application as a function of the engine output, as well as their advantages as compared with other types of reversing gear. To prove the incorrectness of the view that reversible propellers cannot be used for outputs exceeding 200 h.p., the author describes the gear of Theodor Zeise in Altona, Germany, consisting of two reversible propellers for the normal transmission of 1200 h.p., and 1800 effective h.p. as a maximum. Such propellers are now mainly used in submarines for outputs from 100 to 1000 h.p.

ON THE MORE IMPORTANT RESONANCE PHENOMENA AND THEIR EXPERIMENTAL REPRODUCTION (*Über wichtigere Resonanzerscheinungen und deren experimentelle Vorführung*, R. Hartmann-Kempf. *Zeits. für den phys. u. chem. Unterricht*, no. 24, 1911, p. 325, through *Deutsche Mechaniker-Zeitung, no.* 19, p. 200. October 1, 1911. 1½ pp. *e*). Description of apparatus for the experimental reproduction of resonance phenomena, mechanical, accoustical, optical or electrical.

PRESSURE ON THE WALLS IN SILOS AND GAS PRODUCERS (*Wanddruck in Silos und Schachtöfen*, G. Lindner. *Mitteilungen über Forschungsarbeiten auf dem Gebiete des Ingenieurwesens*, no. 124, 1912, p. 1, 32 pp., 42 figs. *te*). Investigation of the pressure on the walls of cylindrical containers, e. g. silos, gas producers, blast furnaces, etc. The theory of earth pressure cannot be applied in such cases directly owing to the small diameter of the structures, under consideration. The author indicates a simple graphical method for the solution of this problem.

CALCULATION OF ARCHED PLATES (*Berechnung gewölbter Platten*, H. Keller. *Mitteilungen über Forschungsarbeiten auf dem Gebiete des Ingenieurwesens*, no. 124, 1912, p. 33. 50 pp., 54 figs. *mt*). A method for calculation and design of bodies with: (*a*) variable thickness, (*b*) variable diameter of arching, or (*c*) hole in the middle. The author solves his differential equations by an approximate "Calculus of small differences."

HEAVY BELT DRIVES FOR ROLLING MILLS (*Über schwere Riemenantriebe für Walzwerksanlagen*, W. Schömburg, *Stahl und Eisen*, vol. 32, no. 40, p. 1655. October 3, 1912. 4 pp. 2 figs. *ce*). The author shows why belts, woven and made of thin leather tanned with oak bark in vacuo, have displaced ropes for heavy drives in rolling mills, and gives data of some tests on the efficiency of such belts. (Cp. The Journal, May 1912, p. 798.)

REVERSAL AND SPEED-CHANGING DEVICES (*Changement de marche et de vitesse, G. R. Revue de mécanique*, vol. 31, nos. 2 and 3, pp. 162 and 267, August 31 and September 30, 1912. 50 pp., 177 figs. *d*). A list of patents, American and foreign, on reversal and speed-changing devices, with drawings and brief descriptions.

TESTING AND ESTIMATION OF LUBRICANTS (*Prüfung und Bewertung der Schmiermittel*, Professor Holde. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, nos. 35 and 36, pp. 1411 and 1460. August 31 and September 7, 1912. 9 pp., 7 figs. *hp*). Paper read by the author before the Lower Rhine Section of the Verein deutscher Ingenieure. A brief historical sketch of the introduction of modern lubricants in Germany and Austria is given, followed by an extensive discussion of the conditions which lubricants must satisfy for particular purposes. There were cases of explosions occurring in compressor cylinders and due to the presence of carbon deposits from the lubricating oil, even though viscous mineral oil containing only a few per cent of neat's-foot-oil was used. It is possible, however, that in such cases proper cooling of the compressor cylinder was neglected, and a violent oxidation by the compressed air took place, or it may have been that the oil was not thick enough or had too low a flash point. Thus, in one case submitted to the Royal Testing Laboratory, explosions were eliminated when for oil having a flash point of 250 deg. cent. (482 deg. fahr.) was substituted one with a flash point of 300 deg. cent. (572 deg. fahr.).

Deposits in the cylinders of steam and large gas engines must not always be credited to the lubricating oil. In the deposits on steam engine cylinders iron particles and particles of iron oxides have been found which may have come from the superheater, while the presence of large grains of sand and metal shavings would indicate that at least some of the deposit is due to the action of the piston which in its rapid motion takes off all the irregularities and impurities of the cylinder surface; there are as a rule more deposits in the case of engines working with superheated steam. In the investigation of the deposits in the mixture valves of a large coke-oven gas engine made at the Royal Testing Laboratory by H. Schlüter, there were found no traces of oil in the deposits which consisted of 2/3 coal tar derivatives, and 1/3 inorganic matter, such as sand, iron particles, etc., indicating that the gas used was not sufficiently clean. On the other hand, oil is often the cause of trouble; thus, in the case of the resinification of an oil used in a turbine it was found that it contained an excessive amount of soapy matter (corresponding to 0.07 per cent ash, while the percentage of ash in clean machine oil must not exceed 0.01). The oil was also found to contain an excessive amount of organic acids, and the soapy matter was probably produced by the interaction between these acids and the iron of the machine.

The author discusses the modern views as to the chemical testing of lubricants, in accordance with the views of Marcusson and Schwartz, and describes the methods and apparatus used in testing lubricants.

Ball Bearings in Mechanics (*Kugellager in der Mechanik*, A. Bauschlicher. *Deutsche Mechaniker-Zeitung*, no. 18, p. 189, September 15, 1912. 5 pp., 4 figs. *dh*). General, partly historical discussion of the design and application of ball bearings, telling how the Deutsche Waffen- und Munitionsfabriken in Berlin, with Professor Stribeck's assistance, have gradually developed the modern form of ball bearings (for Stribeck's formulae see W. C. Unwin, The Elements of Machine Design, London 1909, p. 333). The author makes the following recommendations for putting in ball bearings: (*a*) if two or more ball bearings are used conjointly, only one of them should be fixed tight axially, i. e., the outer race of one bearing should be fixed with practically no play while all the other races of the bearings should have a longitudinal play of 2 to

5 mm (0.078 to 0.196 in.) in accordance with the length of the shaft and its probable expansion; (*b*) as a rule, the inner races are fixed pretty tightly on the journal of the shaft, but heavy one-sided driving-on should be avoided because it expands the race and prevents easy running. The outer race is gently introduced into the bore in the engine frame, or corresponding place, there being no need to have this race fixed in any way since the friction in the bearing is so slight that there is practically no tendency for the outer race to turn with the balls; (*c*) in placing footstep bearings care should be taken that there be no friction between the fixed race and the shaft which passes through it.

The following recommendations are made as to the lubrication of ball bearings and keeping them free from dust: (*a*) only lubricants free from acids should be used, while vegetable oils (e.g. rape-seed oil) and animal fats should be avoided, because they are apt to become rancid and produce rusting of the bearing; (*b*) for high-speed shafts very fluid mineral oils are preferable and more viscous oils for slower shafts; for exceptionally rapidly revolving ball bearings oil bath lubrication should be used, with not too much oil; where the shaft passes through the bearing, the oil level must not reach the shaft since the bearing itself and particularly the cages produce powerful eddies in the oil; (*c*) for the very large number of cases where the shaft passes through the bearing, single or double safety chambers (stuffing boxes in the case of double chambers are recommended); the outer one, or the one away from the bearing, is packed with felt, while the one towards the bearing serves simply for catching the oil, and is provided with a hole for letting the oil flow back on the bearing; (*d*) for the oil bath lubrication of footstep bearings the arrangement shown in Fig. 8 is recommended, since it contains a special outside bush which helps to keep the ball bearing immersed in oil, the latter being kept always at a certain predetermined level.

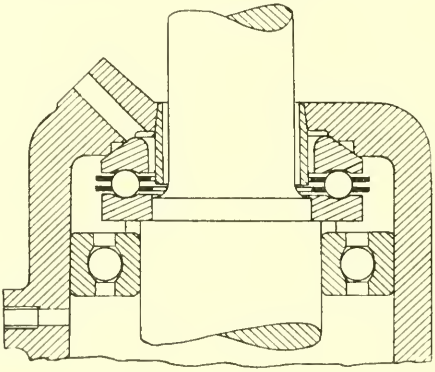

Fig. 8 Oiling Arrangement in a Footstep Ball Bearing

The author points out that although a ball bearing can run for a certain time without any lubrication at all, it becomes heated and wears rapidly away; its consumption of oil is only 1/10 to 1/15 of what an equivalent bearing with sliding friction would take. A further advantage of ball bearings is that should they become worn it is only the bearing that is injured, and not the shaft.

Refrigeration

FLANGE STANDARDS (*Flanschennormalien, Zeits. für die gesamte Kälte-Industrie*, vol. 19, no. 10, p. 198. October 1912. 5 pp., 2 figs., and one table of standard dimensions. *g*). Draft of *Flange Standards for ammonia and sulphurous oxide piping* in refrigerating plants submitted to the German Refrigeration Association by its Committee on Flanges, with a table of proposed dimensions. The Editor calls attention to the fact that the committee based its standards on the standards for flanges of the Verein deutscher Ingenieure for High Pressure Steam of 1900; the Verein, however, lately adopted new flange standards to be known as 1912 Flange Standards for High Pressure Steam.

Shop Management

SEPARATION INTO ELEMENTS AND MEASUREMENT OF A WORKMAN'S EFFORTS, AND THEIR PRACTICAL IMPORTANCE (*Die Zergliederung und Messung von Arbeitsleistung und ihre Bedeutung fur die Praxis*, A. Neuburger. *Werkstattstechnik*, vol. 6, no. 20, p. 529. October 15, 1912. 3 pp., 8 figs., *e*). The French Government granted some time ago a fund of approximately $1000 for the investigation of the problem of scientific management, and A. Imbert of the University of Montpellier has, in connection with this grant, carried on a series of investigations the data of some of which are published in this article. His aim was to obtain exact measurement of the amount of work spent by the workman on his job, and to this end he designed a series of apparatus. As an example may be quoted his study of the process of transporting sacks by a wheelbarrow. The efforts of the workman are divided as follows (Fig. 9A): (*a*) action of the right foot on the axis of the wheel to place the wheelbarrow in an upright position; (*b*) the action of the right hand on the sack producing its tilting; (*c*) simultaneously with the last, action of the left hand on the left hand-bar of the barrow to load the sack on the barrow; (*d*) initial effort applied at the hand-bars to start the barrow; (*e*) effort to push it forward. Each one of these efforts can be measured separately; thus, Fig. 9A gives some idea how the efforts in *b* and *c* are measured: by a suitable system of levers and springs the latter are made to exert in proportion to the effort applied a pressure on a rubber ball filled with air; this drives the air to a special registering apparatus, the effort being later determined from the reading of the registering apparatus. Another device of a similar nature for measuring the work of a filer is shown in Fig. 9B, one of the uses to which it may be applied being the determination of the value of work of apprentices and unskilled laborers. The curves in Fig. 9C present a full picture of the work of a skilled workman on one side, and an apprentice on the other. Curve VII shows how regular and uninterrupted is the motion of the file in the case of a skilled workman, and how with an apprentice there are always dead periods between each forward and backward motion of the file. The curves showing the pressure exerted by the left and right hand are of particular interest. With the skillful workman the pressures of the two hands are in perfect accord, very different from the apprentice's curves.

Steam Engineering

CONTRIBUTION TO THE CRITICISM OF PROCESSES OF FLOW IN NOZZLES AND DISTRIBUTORS (*Zur Kritik der Strömungsvorgängen in Düsen und Leitapparaten*, G. Zerkowitz. *Zeits. für das gesamte Turbinenwesen*, vol. 9, no. 25, p. 394, and no. 26, p. 410. September 10 and 20, 1912. 7 pp., 3 figs. *etA*). Development of the theory of flow of gases through nozzles and distributors, with its application to the case of not perfect gases, i.e. of the equations of Professor Stodola (*Dampfturbinen*, 4th ed.) and Dr. Christlein (cp. The Journal, February 1912, p. 310). Owing to lack of space the mathematical reasoning of the author is omitted, and only a condensed report of the main results is presented here.

Denoting the pressure by p, specific volume by v, velocity of flow of the fluid (wet or superheated steam, gas, etc.) by w, heat content by i, velocity of sound by w_s, weight of the fluid flowing through the nozzle or distributor, in kg per sec., by G, cross-section of nozzle by f, and friction work per kg by R, the author obtains the equation

$$\frac{dw}{wdz} = \frac{w_s^2 \frac{df}{fdz} - \zeta w^2}{w^2 - w_s^2}$$

where z is the length of the path of flow along the axis of the nozzle and ζ coefficient of resistance which the author defines by a special equation in such a manner as to make it independent of the geometric form of the nozzle section. The above equation gives the variation of velocity as a function of the length of the path of flow, independently of the nature of the flowing liquid. and, together with Stodola's equation, permits a full analysis of the process of flow. This equation permits also of the determination in what part of the nozzle the velocity of flow becomes equal to that of sound. With $w = w_s$, the denominator in the above equation becomes equal to zero, but since the acceleration cannot physically become infinite, the numerator in that equation must also become zero, or

$$\frac{df}{fdz} = \zeta$$

which determines the position of the section of the nozzle at which the velocity of flow is equal to that of sound. Further since $\frac{df}{fdz}$ can have only one positive value, the flow can reach the velocity of sound only at a certain cross-section in the interior of a *divergent* tube. The author further points out that for a cylindrical tube, if the coefficient of resistance were equal to zero, both pressure and velocity would be constant while in a tube with a section of minimum cross-section the "acoustic" section might be expected to coincide with that minimum cross-section, though in prac-

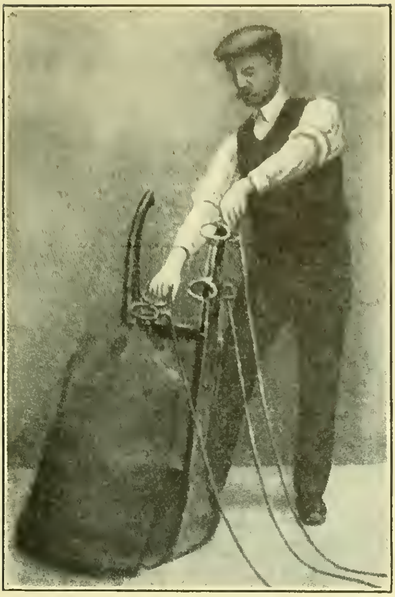

Fig 9a Device for Measuring the Work of a Man Loading a Wheelbarrow

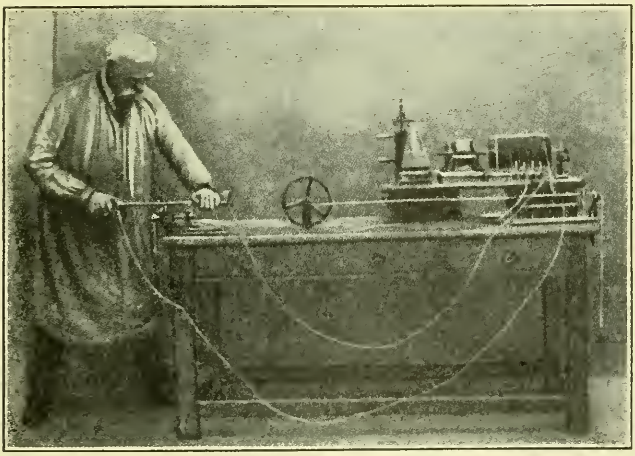

Fig 9b Device for Measuring the Work of a Filer

tice the velocity of flow does not reach that of sound at that section owing to the non-negligible resistances in front of the minimum section observed by Christlein in his Charlottenburg experiments. The difference between the author's formula for the position of the "acoustic" section and that of Lorenz (*Technische Wärmelehre*, 1904) is explained by the fact that while the author deals with velocity of sound, Lorenz uses *adiabatic* velocity of sound. The important distinction between the two is that adiabatic velocity of sound is entirely independent both of the resistances in the piping and the variation of the cross-section along the axis of the pipe, while this cannot be said of the velocity of sound w_s such as the author is using. The author expresses, however, some doubt as to whether either of these sound velocities has really any bearing on the actual process of pressure variations.

By a further investigation of the case of perfect gases the author shows that the sound velocity w_s, which in the case of polytropic expansion coinsides with the velocity of flow, is a function of the exponent of the polytropic expression, and that it depends further not only on the initial state, but also on resistances.

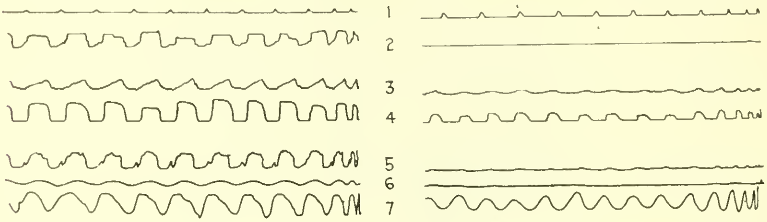

Fig. 9 Devices for Measuring the Efforts of Workman, and Curves showing Work of a Filer

1 Metronome record
2 Horizontal component of the action of the left hand
3 Vertical component of the action of the left hand
4 Horizontal component of the action of the right hand
5 Vertical component of the action of the right hand
6 Variation of pressure on the vice
7 Forward and backward motion of the file

The second part of the article is devoted mainly to a recapitulation of former experiments on the flow of steam through nozzles and on "clearance expansion" (*Spaltexpansion*), with particular regard to the experiments of Christlein above referred to.

Influence of Naphtha on Locomotive Boilers (*Vliyaniye nefti na parovoznye kotly*, B. Artzish. *Bulletin of the Permanent Committee of the Conferences of Representatives of Russian Railroads* (*in Russian*), no. 8, p. 700. August 1912. 3 pp., 1 fig. *p*). It was for some time a common practice on the Russian railroads to *pour naphtha* (10 to 40 lb. at a time) *into locomotive boilers* after washing them out or after the boiler was put up for the first time after coming out of the repair shop. Sometimes naphtha (10 to 15 lb.) was introduced into a boiler which had not been washed for a long time, the purpose being to prevent throwing due to violent formation of steam owing to the dirty condition of the boiler. Generally, however, the naphtha was used because it was supposed to break up the scale in the boiler, and because, carried with the steam into the cylinder, it acted as a *supplementary lubricant for the throttle*, many engineers finding the regular throttle lubrication by the throttle lubricating cup insufficient. It was found that naphtha remains in a boiler some time, and traces of it were found in the water level indicator even after two regular runs.

The author considers such application of naphtha as positively injurious to the boiler, because, as was found in the tests made by Professor Hirsch in Paris, the presence of even a very thin layer of mineral oil on the boiler sheets raises the temperature of the latter considerably, and while the amount of water evaporated was found to fall off about 20 per cent. as compared with the case of the boiler filled with clean water only, the temperature of the boiler sheet was about 180 deg. fahr. higher than in the latter case. This explains why lumps of hard scale separate from the boiler walls more easily in the presence of naphtha, and also why there were so many cases in the Russian boiler practice of fractures of staybolts, cracks in flues, etc., all this being due to overheating produced by the presence of a layer of naphtha on the heating surfaces of the boiler. This is further proved by fractures at the joints of the fire tubes owing to melting of the solder, notwithstanding its being made of two parts copper and one part zinc and melting at 425 deg. cent. (797 deg. fahr.) and higher.

SCALE IN STEAM BOILERS: METHODS OF REMOVING IT AND PREVENTING ITS FORMATION. (*Les incrustations dans les chaudières à vapeur. Remèdes curatifs et préventifs*, Gaillard. *Association des ingénieurs électriciens sortis de l'Institut Electrotechnique Montefiore, Bulletin*, vol. 12, series 3, no. 6, p. 229, June 1912. 70 pp., 25 figs. *gp*). A very complete and interesting discussion and description of various methods of scale prevention and removal, and apparatus used in this connection. Only apparatus of proved practical value or historical interest are considered.

TESTS FOR DETERMINING THE RATING OF HEATING BOILERS (*Versuche zur Ermittlung von Heizkessel-Leistungen*, W. P. *Haustechnische Rundschau*, vol. 17, no. 7, p. 71. October 1, 1912. 3 pp., 4 figs. *ep*). Brief discussion of the proceeding in testing heating boilers based on the respective chapters in the book of de-Grahl: "*Wirtschaftlichkeit der Zentralheizung.*"

SUBSTITUTION OF COMPRESSED AIR FOR STEAM IN OIL BURNING FURNACES (*Substitution de l'air comprimé à la vapeur d'eau dans les foyers à combustible liquide*, Auguste Liévin. *La Technique moderne*, vol. 5, no. 8, p. 294. ½ p., 2 figs. *de*). V. S. Nazaroff, engineer of the Transcaucasian Railroads, Russia, has substituted compressed air for steam in the atomization of liquid fuel, his arrangement as applied to a locomotive being shown in Fig. 10, where C is the compressed air tank, b fuel pipe, and B fuel tank, the fuel used being *mazout*, or residues of rock-oil. The disadvantages which were found in the use of steam for the atomization of liquid fuel consisted in trouble and loss of time in starting; explosions in the furnace produced by the condensation of some of the derivatives of the fuel used forming highly explosive mixtures, and not infrequently burning the firemen; dissociation of steam in the hottest part of the furnace with the formation of a gaseous mixture which on recombining produced noises like pistol shots. All this is said to be eliminated when compressed air is used instead of steam, and further an economy of 38 per cent in the consumption of fuel is secured. In view of the latter statement the Editor calls attention to the data on the use of compressed air for this purpose contained in Mr. Howard Stillman's paper on Locomotive Practice in the Use of Fuel Oils, Trans. Am. Soc. M. E., vol. 33, 1911, p. 49. The article in *La Technique moderne* does not contain full data of any tests.

Concerning Heating Flues in Steam Boilers (*Uber Heizkanäle im Dampfkessel*, P. Koch. *Der praktische Maschinen-Konstrukteur*, vol. 45, no. 21, p. 358. October 10, 1912. 1½ pp. *p*). In passing from pit coal to lignite or vice versa the grate arrangement is sometimes changed while the heating flue cross-section is left as it is. The author shows by a detailed calculation that the *flue cross-section* which may be entirely satisfactory in one case proves to be wrong in another, and prevents the boiler from utilizing the fuel to its best advantage.

Induced Draft Installations (*Saugzuganlagen*, G. Hilliger. *Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 35, no. 40, p. 417. October 4, 1912. 2½ pp. *g*). General discussion of the application of induced draft in steam boiler plants, with some particular references to the Prat, Schwabach and Babcock and Wilcox types.

Counter Current Condensers for Dry Condensation (*Gegenstrom-Kondensatoren für trockene Kondensation*, M. Saalfield. *Der praktische Maschinen-Konstrukteur*, vol. 45, no. 20, p. 334. September 26, 1912. 1 p., 1 fig. and a sheet of drawings. *d*). Description of some German countercurrent condensers, such as are used in the sugar and chemical industries.

A Convenient Control of Steam Consumption of Turbines (*Betriebsmässige Dampfverbrauchskontrolle an Turbinen*, K. Schultze. *Zeits. für das gesamte Turbinenwesen*, vol. 9, no. 28, p. 442 . October 10, 1912. 3 pp., 1 fig. *epA*). Absence of a constant control of steam consumption may lead to heavy losses since one leaky valve or steam connection will, in a year's operation, make quite an appreciable difference in the cost of power. On the other hand the only way to control the steam consumption up to now was by means of steam meters, and their heavy cost makes their constant use prohibitive. The author who had to determine the commercial efficiency of a plant in actual operation, and could rely only on such help as the engineer and fireman of the plant could render, with the engineer having to attend not only to two 1000-kw. turbines, but also to an electric switchboard, had to work out some process that might be applied under such conditions and also under the usual operative conditions in commercial plants. His process was based on the fact that, when $p_2 < 0.58p_1$, or ratio of pressures is below the critical limit already in the first distributor, the steam consumption per second is expressed by the formula

$$G_{sec} = C.F_{min} \cdot \sqrt{\frac{P_1}{v_1}}$$

where $F_{\min}$ is the minimum cross-section of the distributor in square meters, p_1 pressure in atmospheres absolute immediately in front of the distributor, C turbine constant, which for steam temperatures from 250 to 300 deg. cent. (482 to 572 deg. fahr.) varies between 206 and 208, while v_1 is determined from the equation

$$v_1 = 0.0047\frac{T}{p_1} - 0.075\left(\frac{273}{T}\right)^{\frac{3}{10}} + 0.001$$

where T is the absolute steam temperature in front of the distributor.

The author's experiments were made on two 1000-kw. German General Electric Company, Curtis type, turbines with throttling governing, eight nozzles permanently open, and eight more, in two sets of four nozzles each, opened or closed by hand. Readings were taken every hour by the engineer or his assistant, who filled out the first, second, fourth and fifth columns of Table 1, the rest of the work being done mostly by slide rule later in the office. The author points out that a large part of this work can be done by girls, and that it takes only 24 minutes to complete a 24-hour table of steam consumption from the data furnished by the engine room. Values of $\frac{p_1}{v_1}$ are not calculated, but determined from a special chart, such as is given in Fig. 11.

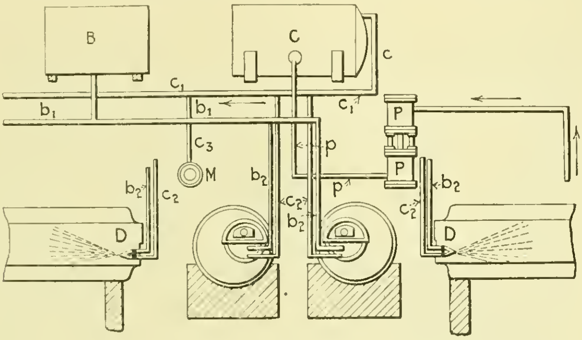

Fig. 10 Nazaroff System of Atomizing Fuel by Compressed Air

It is evident that the correctness of this method depends to a material extent on a correct knowledge of $F_{\min}$, and the author points out that data furnished by the manufacturers are not to be relied on in this respect since contractions in the nozzles are liable to occur either through accumulations of burned oil, or owing to disintegration of the metal itself which partly honeycombs the orifice of the nozzle. This would lead to the calculation indicating a higher consumption of steam than is actually the case, but even that does not deprive the method of its value which lies mainly in indicating fluctuations and thus aiding the discovery of leaks and other irregularities of operation and not in measuring the actual average consumption of steam.

TABLE 1 STEAM CONSUMPTION OF TURBINE NO. 1, FEBRUARY 12, 1912

Time	p_1 Read	p_1 Corrected	t	Nozzles	$\sqrt{\frac{p_1}{v_1}}$	Minutes of Operation	to
7	8.9	9.5	284	9 30	6.50	...	....
8	8.1	8.6	283	I	5.96	...	....
9	7.8	8.2	279	10.35	5.75	210	31.1
10	7.9	8.35	289	II	5.79	65	13.7
				1.15	....	...	....
11	7 1	7.4	274	....	5.24	. .	....
12	8.4	8.9	300	....	6.00	...	....
1	8.3	8.8	290	....	6.04	160	45.0
					Total	435	89.8

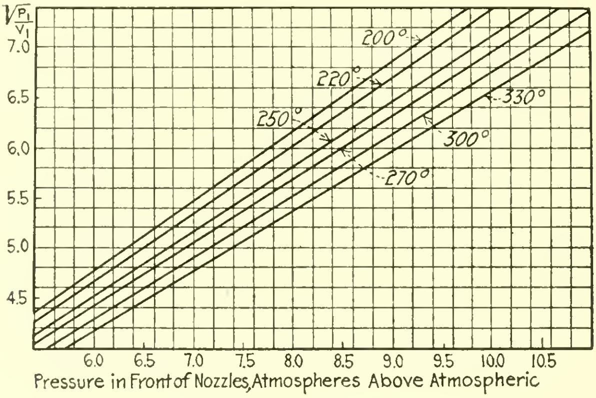

FIG. 11 $\sqrt{\frac{P_1}{V_1}}$ CHART

Strength and Testing of Materials

PORTABLE HYDRAULIC PRESS IN TESTING OF MATERIALS (*Die transportable hydraulische Presse im Materialprüfungswesen*, E. Gebauer. *Tonindustrie-Zeitung*, vol. 36, no. 117, p. 1585. October 3, 1912. 2 pp., 6 figs. *dp*). Description of the portable hydraulic press for testing of materials and its applications. The press installation described consists of the hydraulic press proper with a manometer, a high-pressure hand pump, and beams for placing the press on its foundations. The press is of simple construction, having a long cylinder with a cylindrical piston, with leather packing. The cylinder is made to permit a long useful stroke of the piston, which is of advantage when a long deflection of the piece tested has to be produced, e. g. in the case of a reinforced concrete beam stressed up to rupture. The cylinder is of steel casting, and the whole installation is designed so that pressures up to 300 kg/qcm (4270 lb. per sq. in.) may be easily maintained, without driving the pump too fast in order to replace losses of water escaping through untight places.

Tests of Reinforced Concrete Beams (*Versuche mit Eisenbetonbalken*, C. Bach and O. Graf, *Mitteilungen über Forschungsarbeiten auf dem Gebiete des Ingenieurwesens*, no. 122-123, 1912. *cA*). Fourth part of the publication of the data of tests made by the authors, with numerous tables and illustrations. This is considered to be among the most thorough European investigations of this matter.

Thermodynamics

Determination of the Coefficients of Heat Transmission of Principal Isolating Materials (*Détermination des coefficients de transmission calorifiques des principaux frigorifuges*, M. K. Biquard. *Le Génie Civil*, vol. 61, no. 25, p. 502. October 19, 1912. *e*). From a paper presented by the author at the second French congress of Refrigeration, at Toulouse in September 1912. He found from tests made at the laboratory of the Conservatoire des Arts et Métiers, in Paris, the following values for the conductivity between 0 and 30 deg. cent. (32 and 86 deg. fahr.):

	Specific Weight.		Coefficient of Conductivity.
	Kg/Cbm.	Lb. per Cu. Ft.	
Kapok (vegetable cotton from Java)	15.7	0.94	0.034
Cork, grains	78	4.7	0.040
Cork with casein, dry	148	8.9	0.044
Charcoal	184	11	0.045
Cork with resin	275	16.5	0.052
Corrugated cardboard	..	...	0.052
White asbestos fiber	57	3.42	0.054
Corrugated asbestos cardboard	...	...	0.072
Cork with casein, wet	280	16.8	0.088
Cork with resin	310	18.6	0.076
Compressed asbestos	1240	74.5	0.220
Compressed cellulose	1425	85.5	0.210
Bricks made of asbestos and calcium sulphate	1143	68.5	0.350
Clinker bricks	1400	84	0.400
Soft rock	1470	88	0.430

By interposing air spaces between vertical layers of the isolating material non-conductive walls may be obtained when using even relatively dense materials. The best thickness of the air spaces varies between 8 and 25 mm (0.314 in. to say 1 in.)

Supplementary References

Wazau Dynamometer (*The Journal*, May 1912, p. 806). For detailed description see *Reichelt, Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 35, no. 40, p. 420, October 4, 1912.

Contribution to the Problem of Mechanical Testing of Soft Rubber (*The Journal*, November 1912, p. 1896). Same article reprinted in *Gummi-Zeitung*, vol. 27, no. 1, p. 2, October 4, 1912. Cp. also paper by Prof. Memmler on Mechanical Tests for Rubber in The Rubber Industry, report of the proceedings of the International Rubber Congress, London 1911, p. 351.

GAS POWER SECTION

PRELIMINARY REPORT OF LITERATURE COMMITTEE

(XXII)

ARTICLES IN PERIODICALS[1]

Copper Gas Engine, Some Details of the. *Power*, October 15, 1912. 2 pp., 7 figs. *d.*

Describes a 900-h.p. engine, giving details of design and operation.

Diesel Engined Vessels, Performance of Notable Sea-Going, F. Muller Van Brakel. *International Marine Engineering*, September 1912. 3½ pp., 4 figs. *d.*

Diesel Engines, Motor Ship Eavestone Fitted with Carels. *International Marine Engineering*, October 1912. 2 pp., 3 figs. *d.*

Diesel Engines, Russian High-Speed Marine, J. Rendell Wilson. *International Marine Engineering*, July 1912. 4 pp., 5 figs. *d.*

Diesel Oil Engine, Progress of the, Rudolph Diesel. *Engineering News*, April 4, 1912. 6½ pp., 8 figs.

Read before the Institution of Mechanical Engineers, March 15, 1912.

Junkers Oil Engine, The, F. E. Junge. *Power*, October 29, 1912. 3 pp. *d.*

Details of construction and operation of a 1000-h.p. Junkers oil engine.

Junkers Oil Engine to Marine Work, The Application of the. *International Marine Engineering*, July 1912. 3 pp., 3 figs. *d.*

Luft Vorwärmung bei Verbrennungskraft, A. Nougier. *Die Gasmotorentechnik*, May and June 1912. *dmp.*

On preheating the air for internal-combustion engines.

Oil Fuel, A Great Reservoir of, David White. *Engineering News*, October 17, 1912. 2 pp.

Fuel oil resources of the different shales.

Oil Motor Ship, The Monte Peneds, Another Transatlantic. *Engineering News*, October 24, 1912. ½ p., 1 fig. *d.*

General description of new vessel of the Hamburg-American line for New York and Rio Janeiro service.

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[1] Opinions expressed are those of the reviewer, not of the Society. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *h* historical; *m* mathematical; *p* practical. A rating is occasionally given by the reviewer, as *A*, *B*, *C*. The first installment was given in The Journal for May 1910.

OIL SHIP CHRISTIAN X, NEW HAMBURG-AMERICAN. *Engineering News*, October 3, 1912. 2 pp., 2 figs.

Describes new vessel driven by two 8-cylinder Diesel engines of 2500 total horsepower.

POWER STATION AT KAMATA, JAPAN, GAS-OPERATED. *Engineering*, September 20, 1912. 7 pp., 16 figs., 1 table. *d.*

SULZER-DIESEL ENGINED SHIP FOR NEW YORK–RIO SERVICE, J. Rendell Wilson. *International Marine Engineering*, October 5, 1912. 5 pp., 9 figs. *d.*

L'UTILIZATION DE LA CHALEUR D'ECHAPPMENT, BRABBÉE. *Die Werkstatts Technik*, March 1, 15, April 1, 1912. 5000 words.

Review by Le Mois Scientifique et Industrielle for June 1912.

REPORTS OF MEETINGS

NEW YORK MEETING, NOVEMBER 12

A meeting of the Society was held in the Engineering Societies Building, New York, on November 12, at which a paper on Measuring Efficiency in Manufacturing, was presented by the author, Edward B. Passano. Mr. Passano advanced a theory that all profit should be treated as an item of expense. All loss, through inefficiency, is a potential or a positive profit to be made in enterprise. A reduction in expense of business increases the actual profit and increases the efficiency. This method keeps constantly before the management the actual value of each unit of production to the organization, and the loss through inefficiency which is a potential or possible profit. When a new method is being tried out, such a system will show the benefits or losses which are expressed in actual profit or loss as the work progresses.

The paper was discussed by Harrington Emerson, H. M. Rowe, president H. M. Rowe Company, Baltimore, J. M. Foster, Leroy Tabor, Edward L. Suffern, public accountant, of New York, Chas. B. Going, Wm. Kent, Howard F. Turrill, of Patterson & Ridgway, New York, F. A. Waldron, F. W. Miller, H. L. Gantt, F. B. Gilbreth, C. Bowyer Vaux, of the Wistar Institute, Philadelphia, and A. Hamilton Church, consulting engineer, of New York.

NEW HAVEN MEETING, NOVEMBER 13

A meeting of the Society was held in New Haven on November 13, 1912, with afternoon and evening sessions. At the first session which opened in the Mason Laboratory of Mechanical Engineering, Sheffield Scientific School, at three o'clock, E. S. Cooley presided, and two papers on the Transportation of Materials in Manufacturing Plants were read by Herbert L. Seward, Jun. Am. Soc. M. E., of the Sheffield Scientific School, and F. C. Bennett of the Stanley Works, New Britain. The former of these was illustrated by lantern slides. These were discussed by C. D. Rice, Harwood Frost, W. S. Huson, Prof. L. P. Breckenridge, E. S. Cooley, Arthur Brewer, C. F. Hutchings, Calvin W. Rice and Mr. Goldstein, of the Link-Belt Company, Philadelphia. A feature of this session was a large collection of catalogues, photographs and books illustrating all types of conveying machinery, and an extensive bibliography of books and articles on the subject, prepared by the library of the Engineering Societies. At the conclusion of this session dinner was served at the Yale Dining Club.

The evening session opened at eight o'clock in the Mason Laboratory and Calvin W. Rice, Secretary of the Society, was introduced by Prof. Charles F. Scott. Mr. Rice spoke on the work of the Society, particularly of the local branches, and of the coming Annual Meeting and the Joint Meeting to be held in Germany next summer.

C. E. Booth, chairman of the Yale Student Branch, who followed, spoke briefly of the work planned by the students, including an interesting program.

The subject of the evening, Industrial Education, was then presented in two papers, both illustrated by lantern slides, one by Professor Breckenridge and the other by F. J. Trinder, principal of the New Haven State Education Shop, describing the work of the two Connecticut trade schools at New Britain and Bridgeport. The subject was of particular interest locally, as it is now under consideration in New Haven with reference to the public school system.

The entire meeting was largely attended, more than 150 being present, some of whom came from a considerable distance, showing the wide interest aroused by the occasion.

BOSTON MEETING, NOVEMBER 15

At the November meeting of the Society in Boston, held in Edison Hall on Friday, the 15th, Prof. E. F. Miller presided. The paper of the evening, Dry Rot in Timber Used in Slow Burning Construction, was presented by the author, Frederick J. Hoxie.

On account of the extent of the subject Mr. Hoxie confined himfined himself to the types of fungus growth, roughly classed as dry rot, and to their effect on the timber generally used in mill construction, and gave special attention to yellow pine. In the beginning he called attention to the fact that as a result of the continued demand the quality of timber available for mill construction is constantly growing poorer, also to the difficulty of making specifications which should absolutely insure the desired quality of timber being obtained by reason of the wide difference in the character of woods covered by the same commercial designation. He recommended grading of lumber according to the number of annual grain bands and the specific gravity, rather than by the usual botanical designations. While dry rot is not as common as other less troublesome fungi, and is not found as extensively here perhaps as abroad, it was pointed out that it was exceedingly dangerous because of its tendency to develop unobserved in the interior of sticks and at points where they were built into the masonry. The relative susceptibility of heart and sap wood to this infection was touched upon, also effect of the proportion of rosin. It was shown how the infection was spread to sound lumber, both through faults in piling the lumber in yards and also when assembled in the building. The author called attention to some steps which might be taken to check the infection when once started, and suggested that usually when infection became noticeable, injury had already gone so far as seriously to weaken the structure. Preventive measures were more strongly recommended and several processes of treatment of lumber were discussed.

At close of the written paper Mr. Hoxie showed a large number of slides illustrating the development of the fungus in lumber and the contributory part played by the rot in ruined structures. Several of these slides were from micro-photographs showing very clearly the spread of the fungus through the structure of the stick.

Charles T. Main. Mem. Am. Soc. M. E., referred to the mischievous effects of painting timbers in buildings before they had opportunity to become thoroughly seasoned. and discussed at length several processes for preserving lumber. Further discussion was offered by Dwight Seabury, Mem. Am. Soc. M. E., R. A. Hale, member of the Boston Society of Civil Engineers, and others.

STUDENT BRANCHES

ARMOUR INSTITUTE

A meeting of the Armour Student Branch was held on November 6, at which Mr. Roesch, a member of the faculty, presented a paper on Carbureters and Mixing Valves. covering mixing appliances for gas engines and a large number of carbureters adapted to automobile work. In addition to lantern slides, Mr. Roesch showed a number of carbureters, parts of which were cut away. The paper was discussed by E. R. Burley, H. A. Bohlander, E. S. Libby, H. R. Kuehn, Chas. Kehr, and D. A. Drozeski.

COLUMBIA UNIVERSITY

At a meeting of the Columbia University Student Branch on October 30 reports on power plant experience were read by L. Grossbaum and Howard Swallow, and practical work in the shops and drafting rooms of representative manufacturing establishments was described by J. Stuart and Mr. Senkstaken.

LEHIGH UNIVERSITY

A paper on Natural Gas Investigations in the Bureau of Mines, by George A. Burrell, was read at a meeting of the Mechanical Engineering Society of Lehigh University on October 22, and discussed by Mr. Mart and Professor De Schweinitz; and another on the Fatigue Failure of Metals, by G. B. Upton and G. W. Lewis, was read and discussed by Mr. Spencer.

OHIO STATE UNIVERSITY

The members of the Ohio State University Student Branch were guests of Prof. F. E. Sanborn, Mem. Am. Soc. M. E., on October 21, when the regular meeting of the organization was held. Prof. Wm. T. Magruder, Mem. Am. Soc. M. E., was unanimously chosen Honorary Chairman of the branch. An illustrated account of his trip through England and Scotland was then given by Professor Magruder, and a general discussion followed.

PENNSYLVANIA STATE COLLEGE

The following officers have been elected for the year by the Pennsylvania State College Student Branch: President, Jonas F. Blank, Vice-President, Wm. E. Evans, Secretary, Guy W. Barger, and Treasurer, Gordon R. Reese. At the first meeting of the branch for the season, held on October 18, addresses were made by Dean J. P. Jackson. and Professors Hugo Diemer, J. A. Moyer, A. J. Wood and J. A. Mease, Members Am. Soc. M. E.

POLYTECHNIC INSTITUTE OF BROOKLYN

On November 12, an illustrated lecture on the 1910 Expedition to Mt. McKinley by J. H. Cuntz was given before the Polytechnic Student Section. A discussion followed in which Henry Priester, B. L. Heustis, Mr. Adler and others participated.

SIBLEY COLLEGE

At the first meeting of the Sibley College Student Branch on October 10 Prof. D. S. Kimball, Mem. Am. Soc. M. E., spoke on Pittsburgh and Its Industries, before a large audience. Dr. William Kent, Mem. Am. Soc. M. E., addressed the meeting of October 30 on the Value of Affiliation and also on Educational Efficiency.

UNIVERSITY OF ARKANSAS

The following officers were elected on October 15 at the University of Kansas for the year 1912-1913: Honorary Chairman, B. N. Wilson, President, Minto McGill, Vice-President, J. G. Buerkle, and Secretary, Claude Bethel. An impromptu debate on the subject, Resolved that the reciprocating engine is more efficient than the steam turbine for sizes below 500 h.p., followed.

At a meeting on November 5, Minto McGill opened a discussion on high-carbon steel, in which various members of the branch participated.

UNIVERSITY OF CINCINNATI

At a meeting of the University of Cincinnati Student Branch held on November 1, in which the Student Branch of the American Institute of Electrical Engineers joined, Mr. Fishback, sales maanger of the Electrical Controller Company of Cleveland, gave an illustrated lecture on the Handling of Iron and Steel by Electrical Methods.

UNIVERSITY OF ILLINOIS

Charles R. Velzy (Student 1914) addressed a meeting of the University of Illinois Student Branch on October 26, on the Life of James Watt, which was illustrated with lantern slides. On November 12, Mr. A. S. Buyers of the mechanical engineering department talked on the Theory and Use of the Slide Rule. A general discussion followed.

UNIVERSITY OF KANSAS

A meeting of the University of Kansas Student Section was held on October 24, at which Prof. C. M. Young, of the Mining Engineering Department, read a paper on Explosives. Nature and Composition, which the members joined in discussing. Two papers, The Production of Petroleum Oils, by W. H. Tangeman (Student 1910), and Oil Burners, by Earl Carson (Student 1912), were presented at the meeting of November 7 and discussed by those present.

UNIVERSITY OF NEBRASKA

A paper on Manometer and Piezometer Tubes, by Prof. J. D. Hoffman, Mem. Am. Soc. M. E., was presented at a meeting of the University of Nebraska Student Branch on November 5, and was discussed by L. C. Lichty and B. E. Morley.

EMPLOYMENT BULLETIN

The Society considers it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is pleased to receive requests both for positions and for men. Notices are not repeated except upon special request. Names and records, however, are kept on the current office list three months, and if desired must be renewed at the end of such period. Copy for the Bulletin must be in hand before the 12th of the month. The list of "men available" is made up from members of the Society and good men not members. Information will be sent upon application.

POSITIONS AVAILABLE

0223 Department manager wanted in a large works near New York, making light metal products of high grade, to organize, direct and control the movement of product through the plant, including regulation of stock and shipment of orders. Experience in mechanical and electrical engineering preferred, and training in up-to-date methods of organization, system and management essential. Give age, education, experience and references. Apply through Society.

0224 Works superintendent for plant employing 400 men, manufacturers of Corliss engines and other heavy machinery. Entire charge of plant will be given competent superintendent. Apply through Society.

0225 Commercial engineers for lubricating oil business. Should be technically inclined, with good selling experience and acquaintance with manufacturers. New York State and Connecticut.

0226 Assistant to purchasing agent, to take care of engineering specifications and contracts and act as consulting engineer; position calls for general experience in mechanical engineering and in all matters relating to construction. Location New York State.

0227 Competent engineer as assistant to chief engineer in charge of power plants, regular steam and electrical equipment and large amount of both absorption and compression types of refrigerating machinery. Thoroughly competent man. Location New York State.

0228 Expert on design of automatic machinery, who has been successful along this line.

0229 Young man, technical education, few years' practical experience in connection with boiler room work, for testing and investigating along lines of perfecting combustion.

0230 Tool designer, experienced man on modern cutting shapes and heat treatment, capable of maintaining tool equipment in most economical manner and supervising tool room work. State fully experience, present employment and salary.

0231 High class draftsman and designer of factory layout and special machinery, to carry out ideas in highly confidential capacity. Prefer man of at least 15 years' experience in shop practice and designing. Apply through the Society.

0232 Competent man to take charge of experimental department working in touch with the manufacturing department in view of devising and remodeling all types of harvesting machinery, including tillage implements. One who has had large experience in the field preferred, or a mechanical engineer with extensive shop experience in designing and constructing various types of machinery and capable as an organizer and superintendent.

0233 Designer capable of handling draftsmen and producing results. Certain amount of shop experience would be an advantage, with experience in industrial plant work. State salary expected and give complete references. Apply through the Society.

0234 Mechanical engineer having experience in designing and knowledge of locomotive work, one preferred who has specialized in stoker design and manufacture.

0235 Chief of bureau of gas and lighting for eastern city. One with the education of a chemical engineer with training and experience of a business man preferred. Apply through the Society.

MEN AVAILABLE

538 Life member, 35 years' experience in designing cutting machines and all apparatus including extracting and evaporating devices for putting raw material into marketable products of dye and farming woods, desires position. Has installed plants for economical utilization of spent wood for generating steam used in the plant.

539 Member, age 34, wide experience in design, construction and supervision of power plants, piping systems and general factory engineering, desires position as works engineer. Power plant economy and efficiency a specialty.

540 Mechanical engineer or superintendent for large corporation; experience along broad mechanical and steam power lines; executive, with ability to handle large number of men, or will consider proposition to act as consulting engineer.

541 Assistant general manager, at present employed in this capacity, technical graduate, 16 years' experience in combined manufacturing and engineering sales work, seeks responsible executive position, preferably with a concern manufacturing a standardized product rather than special work.

542 Junior, 30 years of age, Stevens graduate, desires position as publicity manager for concern manufacturing engineering machinery, or position of similar nature. Has had three years' experience, editor of trade paper, also general business experience.

543 Mechanical engineer, experienced in design, construction and maintenance, competent to act as works engineer.

544 Member, 23 years' experience on rolling mills, miscellaneous steel works, hydraulic and special machinery, desires responsible position. Competent to take entire charge of engineering department in any works building the above lines of machinery.

545 Member with 15 years' broad mechanical experience, at present employed as sales engineer, desires change. Qualified for assistant superintendent, chief draftsman, estimator or sales engineer.

546 Member, technical graduate, age 37, married, wants position as manager. At present employed. Has had practical, manufacturing and sales experience, both as superintendent and as sales manager. Has thorough knowledge of production, costs and manufacturing processes. Would prefer nominal salary with percentage of profits.

547 Mechanical engineer, age 32, married, with initiative, resourcefulness and executive ability; technical education and experience in design, construction and erection, estimating, valuations, purchases and testing; investigations and reports, general and special machinery and industrial plant equipment, special proficiency in internal combustion motors. Prefers location in moderate sized Eastern city. Salary $3000.

548 Junior member, 29, married, twelve years' experience in shop drawingroom and foundry, five years in executive positions, wishes position as assistant superintendent or engineer. Good organizer, designer of automatic machinery and foundry equipment for large production. Thoroughly acquainted with operation of grey iron and malleable foundries. Salary $1800.

549 Mechanical engineer, member, university graduate, 25 years' experience in the building, erection and managing of sugar and chemical plants, designing and building special apparatus, vacuum and condensing plants, power plants, etc., will consider position as superintendent, managing engineer of industrial plants, or take charge of branch office for equipment of such plants.

550 Junior member, technical graduate, experience in fire protection and operating department of large light and power company, desires position with manufacturing concern or consulting engineer.

551 Technical graduate, M.E. and C.E. Wishes to take charge of machine design or descriptive geometry departments of a university or technical school. Has been professor for a number of years in machine design and descriptive geometry, and is author of recent text-books on these subjects. Would accept position as mechanical or research engineer in manufacturing plant, consulting and testing work. At present employed.

552 Junior member, with three years' experience in heating and ventilating work and power plant installations would associate with architect's or consulting engineer's office, taking charge of some line of work. Graduate of a leading Eastern technical school. At present employed.

553 Junior, age 30 years, technical graduate, with experience in design, construction, operation and maintenance of power plants and substations, and in testing electrical and other machinery, desires responsible position with large manufacturing plant or with contractors doing power plant or hydroelectric work. Salary $175-$200 per month.

554 Mechanical engineer, extensive practical shop and commercial experience, expert in up-to-date manufacture of interchangeable machinery, can reduce costs and increase production. At present employed.

555 Associate, age 35, desires position in Philadelphia or vicinity. At

present employed as checker in drawingroom. Experienced in civil, structural and mechanical lines.

556 Mechanical engineer with general engineering, construction, railroad and factory experience desires position. Recently engaged in valuation work.

557 Technical graduate experienced in selling, sales supervision and organization desires executive sales position. Competent in design, operation and sale of power plant equipment, including water purification.

558 Member, graduate mechanical engineer, age 39, married, desires position as superintendent, plant engineer or purchasing agent. Has technical and commercial knowledge in machine shop and foundry practice. At present employed. Location near New York. Salary $3500.

559 Member, technical graduate, B.S. and M.E., at present employed. Lighting, heating and ventilation preferred, but not insisted upon.

ACCESSIONS TO THE LIBRARY

WITH COMMENTS BY THE LIBRARIAN

This list includes only accessions to the library of this Society. Lists of accessions to the libraries of the A. I. E. E. and A. I. M. E. can be secured on request from Calvin W. Rice, Secretary, Am. Soc. M. E.

AMERICAN CERAMIC SOCIETY. Trans. vol. 14. *Columbus, 1912.* Gift of society.

AMERICAN SOCIETY OF CIVIL ENGINEERS. Index to Trans. vols. 1-74, 1867-1911. *New York, 1911.*

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Trans. vol. 33, 1911. *New York, 1912.*

AMERICAN WOOD PRESERVERS' ASSOCIATION. Pro. 8th Annual Meeting, 1912. *Baltimore, 1912.* Gift of association.

AMERIKANISCHE HÜTTENWERKE, Peter Eyermann. *Wein, 1912.* Gift of author.

BRITISH FIRE PREVENTION COMMITTEE. Red Book no. 169. *London, 1912.*

CAMBRIDGE (MASS.) WATER BOARD. Report for the year ending March 31, 1912. *Boston, 1912.* Gift of board.

CARELS-DIESEL ENGINE. Gift of W. R. Haynie.

CONCRETE HIGHWAYS. *Philadelphia, 1912.* Gift of Association of American Portland Cement Manufacturers.

ELECTRIC LIGHTING AND MISCELLANEOUS APPLICATIONS OF ELECTRICITY, Wm. S. Franklin. *New York, The Macmillan Co., 1912.*

This volume is particularly interesting in that although it is a text book for the use of students, the first chapter covers installation and operation costs and the selling price of electrical energy, thus entering in the very beginning into a discussion of the most important part of electrical engineering. Professor Franklin's book, which is intended to be a companion volume to his Dynamos and Motors, is evidently intended to be used by his students in Lehigh University.

ELECTRIC POWER FROM THE MISSISSIPPI RIVER. Bull. Nos. 1-7, 1911-1912. *Keokuk, 1911-12.* Gift of B. H. Parsons.

ELEMENTARY MACHINE DRAWING AND DESIGN. Wm. C. Marshall. *New York, McGraw-Hill Book Co., 1912.*

Professor Marshall's book has been planned for students in engineering who have had a previous course in orthographic projection, but who have not yet become acquainted with simple machine elements. It is primarily a text book for the students in the Sheffield Scientific School of Yale University, but will undoubtedly find large acceptance in other engineering schools.

ENERGY AND VELOCITY DIAGRAMS OF LARGE GAS ENGINES, THEIR USE AND LAY-OUT, Paul L. Joslyn. *Cincinnati, Gas Engine Publishing Co., 1912.*

In this book the author gives the methods of laying out energy and velocity diagrams for large engines operating on blast furnace, producer or natural gas, with instructions as to their use, etc. The data given are the result of actual designing of this character on some of the largest engines built in America and Europe and will be found of inestimable advantage to the designer working on engines of this character.

DIE ENTROPIE-DIAGRAMME DER VERBRENNUNGSMOTOREN EINSCHLIESSLICH DER GASTURBINE, P. Ostertag. *Berlin, 1912.*

DIE EXISTENZ DER MOLEKÜLE. EXPERIMENTELLE STUDIEN, The. Svedberg. *Leipzig, 1912.*

FLOOD WALL AND MODERN WHARF. Location and Description. *Pittsburgh, 1912.* Gift of Flood Commission of Pittsburgh.

FORSCHERARBEITEN AUF DEM GEBIETE DES EISENBETONS. Pt. 20. *Berlin, 1912.*

GRINDING WHEELS. *Hartford, 1912.* Gift of Travelers Insurance Co.

HETCH HETCHY WATER SUPPLY FOR SAN FRANCISCO, 1912. REPORT, John R. Freeman. *San Francisco, 1912.* Gift of John R. Freeman.

IOWA STREET AND INTERURBAN RAILWAY ASSOCIATION. Pro. 8th Annual Convention, 1911. *Davenport, 1912.* Gift of association.

JAHRBUCH DER MOTORLUFTSCHIFF STUDIENGESELLSCHAFT. 1907-1908, 1909-1910. *Berlin.*

MANCHESTER STEAM USERS' ASSOCIATION. Memorandum by Chief Engineers 1911. *Manchester, 1912.* Gift of association.

MARKET STREET EXTENSION RAPID TRANSIT TUNNEL, UNDER TWIN PEAKS RIDGE, REPORT ON, Bion J. Arnold. Preliminary Report No. 8, submitted October 7, 1912. *1912.* Gift of author.

MECHANICAL ENGINEERING LABORATORY MANUAL, Earl B. Smith. Ed. 2. *Philadelphia, Drexel Institute Press, 1912.* Gift of author.

The author states that his endeavor has been to give a few experiments along different lines that would serve to the best advantage the purpose of educating young engineers, and not to give them a great store of information but to teach them principles. Experiments on the strength of materials and on cement testing has been left for another volume. Some of the experiments contained in the first edition, which was issued in 1909, have been omitted, several have been revised and a few added to the second edition.

MECHANISM, R. McA. Keown. *New York, McGraw-Hill Book Co., 1912.*

METROPOLITAN SEWERAGE COMMISSION OF NEW YORK. Report 1912. *New York, 1912.* Gift of commission.

MODERNE TECHNIK, H. Blücher. 2 vols. *Leipzig, 1912.*

MODERNE WINDTURBINEN, Otto Stertz. *Leipzig, 1912.*

MOTEURS À DEUX TEMPS, L. Ventou-Duclaux. *Paris, 1912.*

NATIONAL MACHINE TOOL BUILDERS' ASSOCIATION. Official Report. May 1911, October 1911, Atlantic City, New York 1911.

NATIONAL RIVERS AND HARBORS CONGRESS. Pro. 8th Annual Convention, 1911. *Cincinnati, 1912.*

NEW YORK STATE EDUCATION DEPARTMENT. Souvenir of the Dedication of the New York State Education Building, October 15-17, 1912. *Albany, 1912.* Gift of department.

DER PORTLAND CEMENT UND SEINE ANWENDUNGEN IM BAUWESEN, F. W. Büsing and C. Schumann. *Berlin, 1912.*

PRACTICABILITY, UTILITY AND BENEFIT OF RAILROADS. Inland Transit, N. W. Cundy. Ed. 2.

PROCURING AND MAINTAINING FOREIGN PATENTS. *Washington, 1910.* Gift of Alexander and Dowell.

THE RUDDER. vol. 16, 1905. *New York, 1905.*

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7, nos. 1-12; vol. 8, nos. 1-12; vol. 9, nos. 1-12; vol. 10, nos. 1-12; vol. 11, nos. 1-5, 7-12; vol. 12, nos. 1-12; vol. 13, nos. 1-10, 12; vol. 14, nos. 1-12; vol. 15, nos. 1-2, 4-5, 7, 9-12; vol. 16, nos. 1-12. *St. Louis, 1896-1912.* Gift of St. Louis Railway Club.

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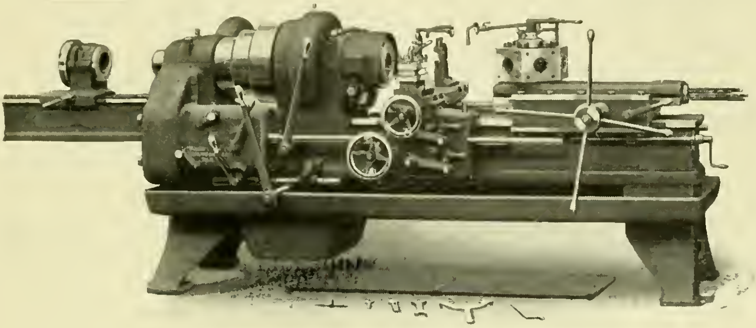

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Multi-stop and Double Turners

FIG. 1 illustrates the advantage of the *double stop* for each position of the turret, and the *double* adjustment of each turner. This piece has six finished diameters and six shoulders, and is turned by only three turners, which occupy only three positions on the turret. This not only leaves the remaining positions free for other tools, but it saves the operator the time and energy required to run the turret slide back each time.

All this is obtained without complication, and without introducing any features that are annoying when not in use.

In addition to the double stop for each of the six positions of the turret, we have an extra stop, consisting of a pin which may be dropped into any one of the six holes at the rear of the turret slide. This makes it possible to borrow five extra stops for any one of the tools, and gives to this tool seven length or shoulder stops, and leaves one stop for each of the remaining tools.

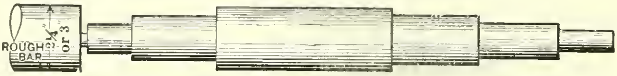

FIG. 1

The illustrations, Figs. 2 and 3, give examples of what one tool can do in this machine on chuck work, when we take advantage of the seven length stops and the seven shoulder stops of the cross-feed head.

Of course, in general practice three or four stops for one tool are all that will be needed, but since the modern cutting steels have greater durability, there is nothing lost by giving each tool all the work it can do.

Outer face and all shoulders and diameters accurately finished to independent stops by one tool. When roughing and finishing cuts are required, the roughing tool can be set near enough to use the same stops that are accurately set for the finishing tool. When an extra tool is used to give a roughing cut it is set as indicated by dotted lines in Figs. 2 and 3.

We find it difficult to illustrate all of the classes of work that can be turned out by this machine, but a little thought will suggest

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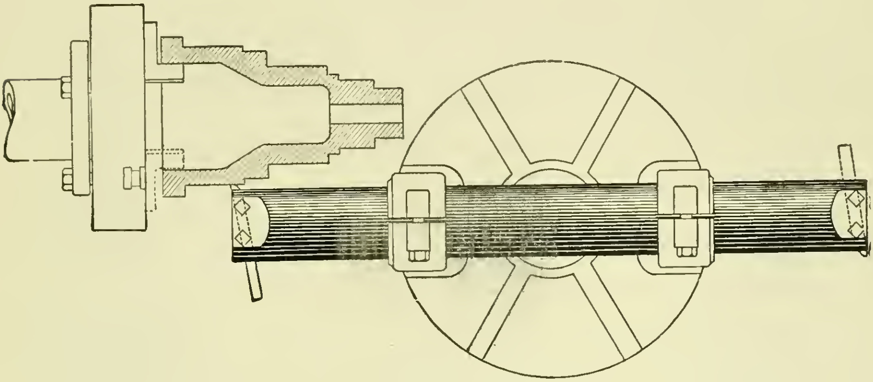

FIG. 2

many forms that may be readily handled in bar and chucking work, both steel and iron, on account of the many provisions for bringing both turret and cross slide up to fixed stops; either by power feed or by hand.

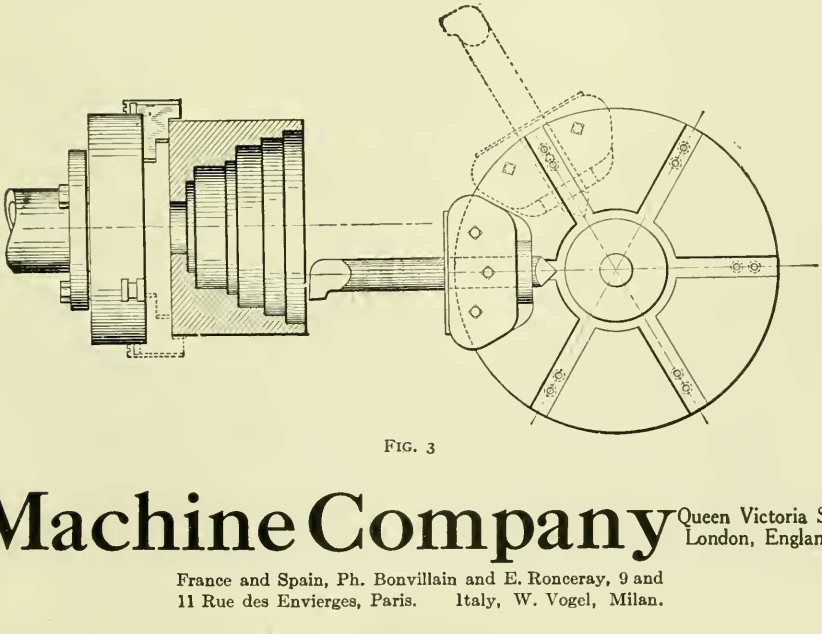

FIG. 3

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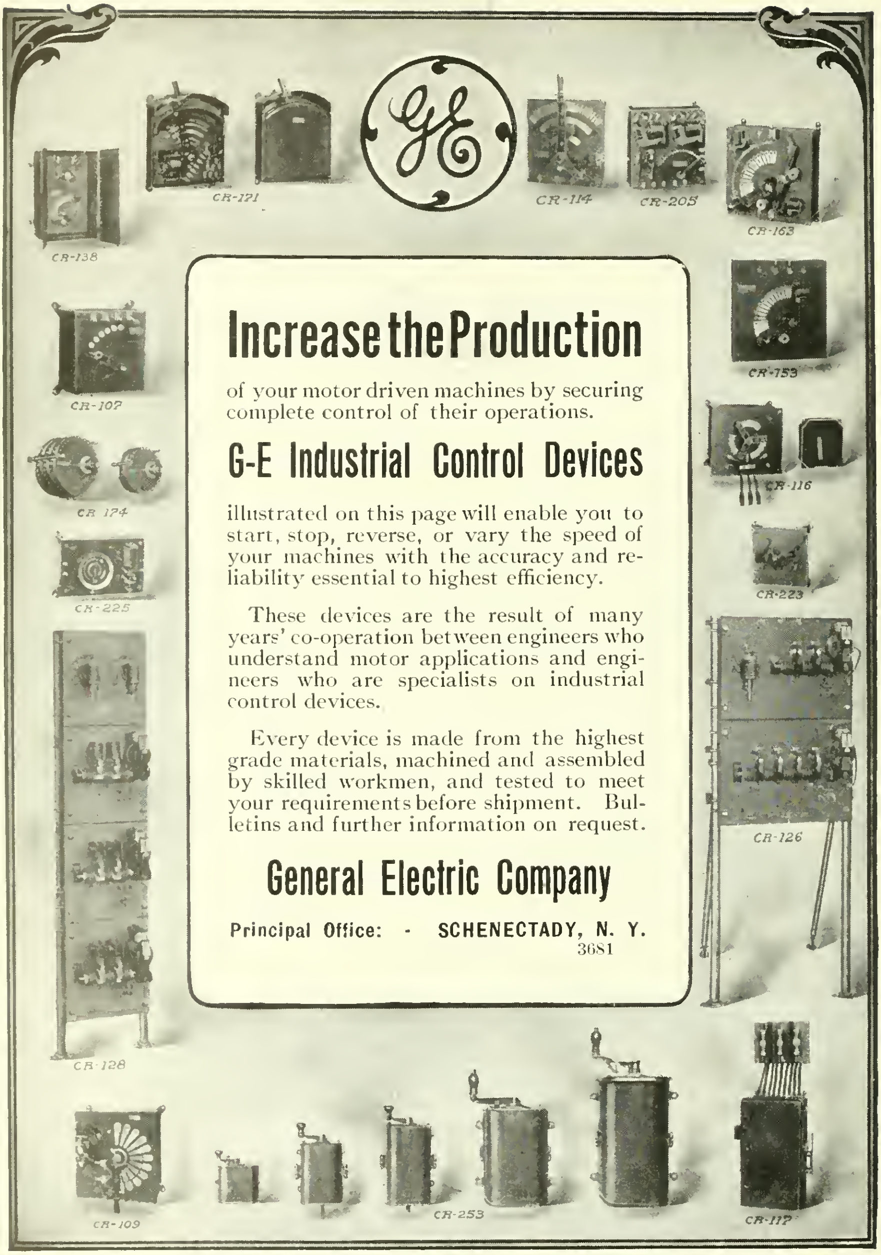

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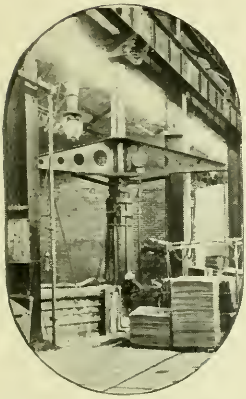

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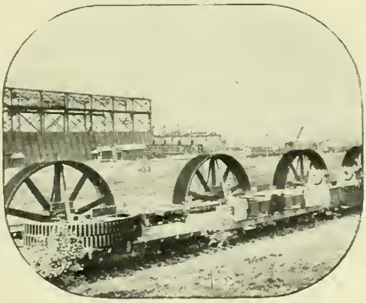

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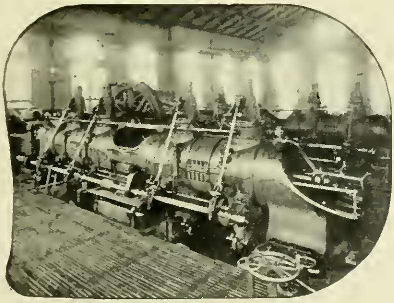

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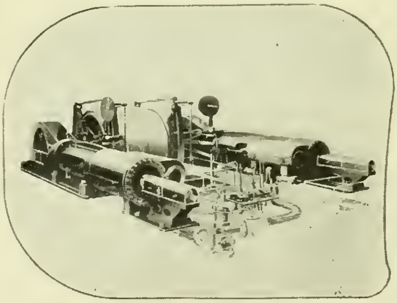

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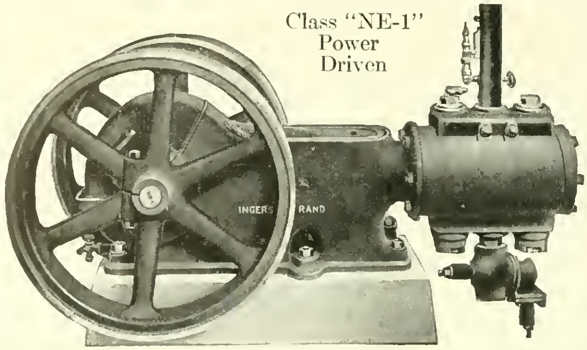

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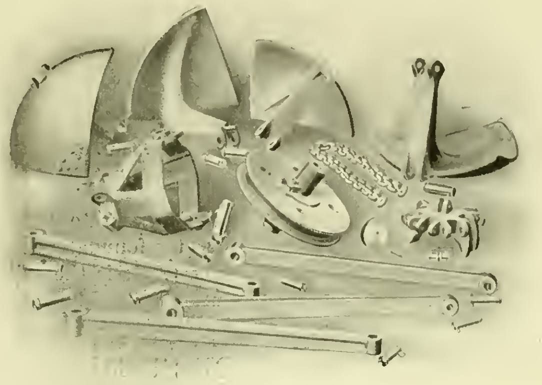

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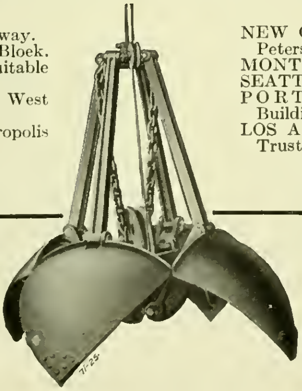

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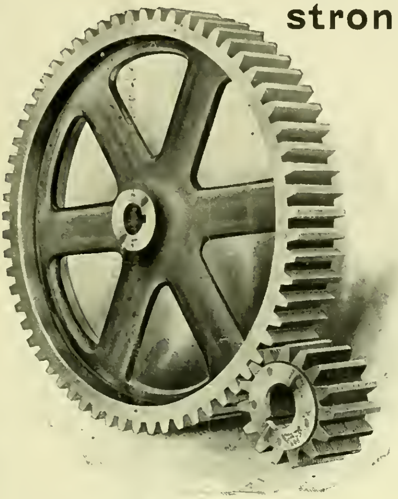

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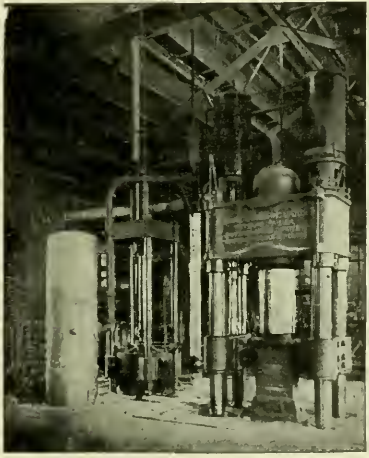

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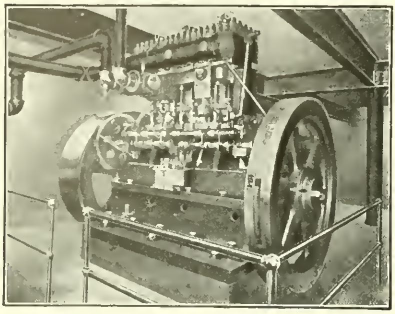

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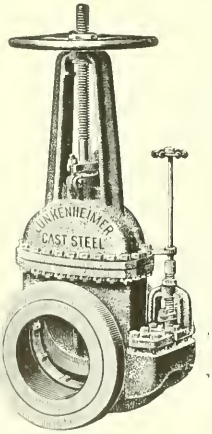

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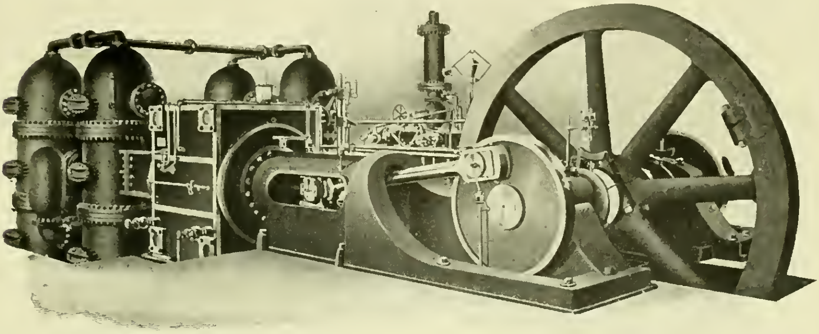

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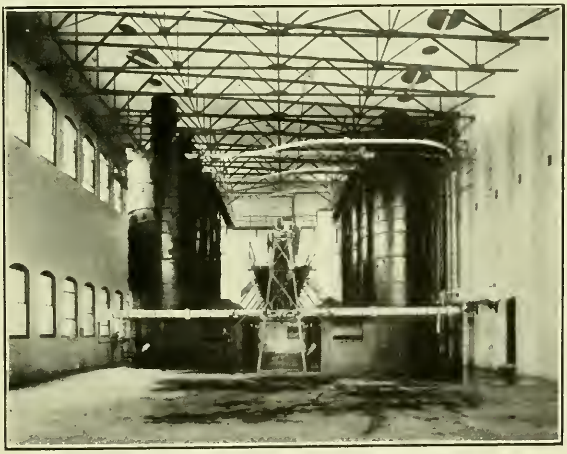

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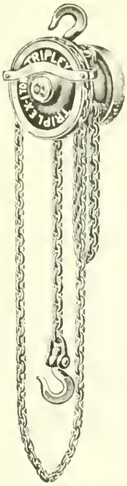

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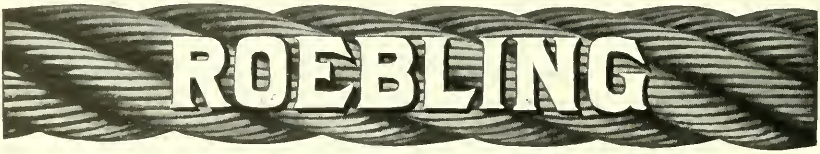

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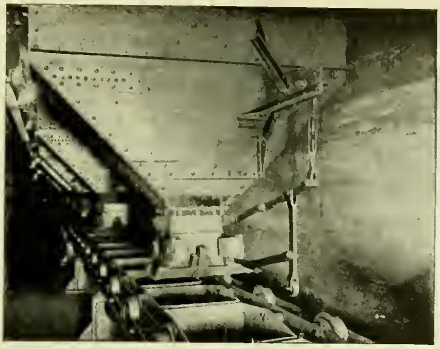

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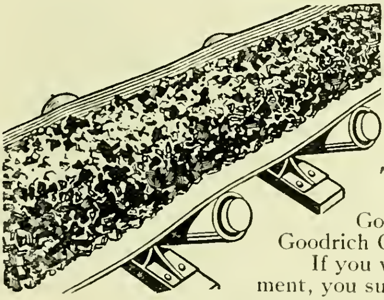

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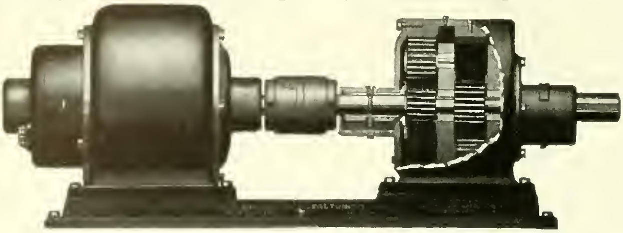

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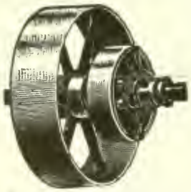

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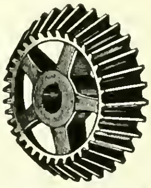

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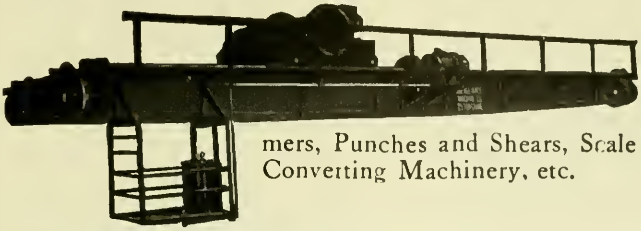

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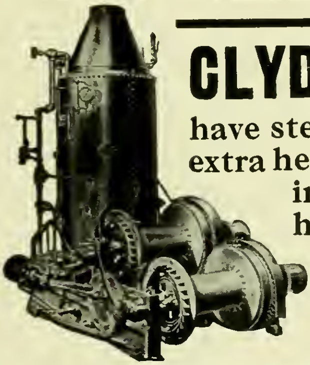

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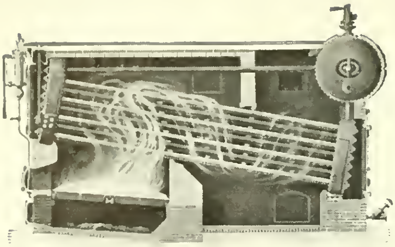

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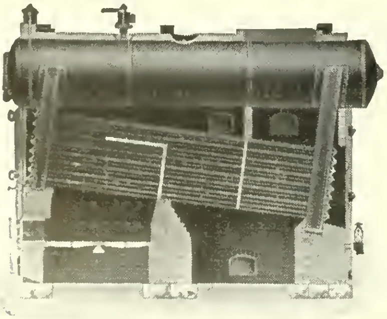

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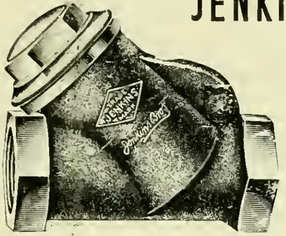

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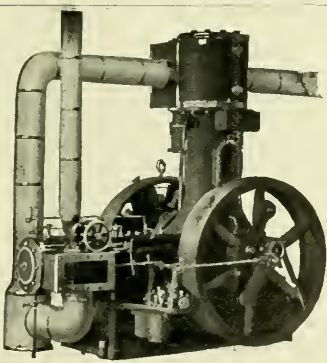

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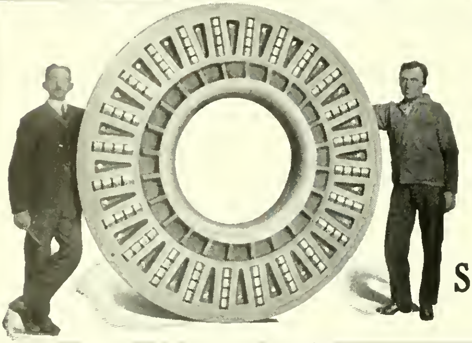

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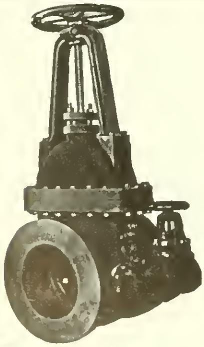

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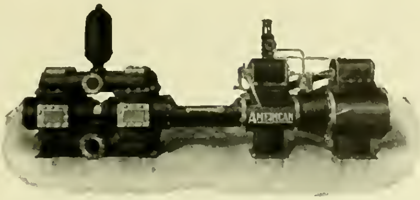

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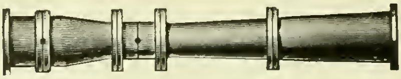

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